PRODUCT ENGINEERING

VOLUME 7 NUMBER 2

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In this Issue

We commemorate the fiftieth anniversary of the discovery of the electrolytic reduction of aluminum by Charles Martin Hall, on February 23, 1886. Hence the three feature articles relating to aluminum.

Coming Issues

The largest single expense factor under the sole control of the engineering department is the cost of keeping in stock an adequate supply of raw materials, semi-finished and finished parts. By reducing the number of sizes required, the inventory costs can be reduced correspondingly. In the March number R. E. Hellmund and D. F. Miner of Westinghouse will explain how to do it.

Flexible bellows are finding more and more use not only as temperature sensitive elements but also as flexible seals. Standard types available, scope and limitations and typical applications will be set forth in a feature article in the March number.

Other feature articles in forthcoming issues include "The Evolution of the Built-In Motor" by R. H. Rogers of General Electric, "The Flow of Metals in Stampings" by Earl C. Booth of Noblitt-Sparks, "Typical Hydraulic Applications," and "Finishing Specifications."

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CONTENTS for FEBRUARY, 1936

REDESIGNING TO USE ALUMINUM 42
GEORGE W. DEBELL—How the difference between the elastic modulus of structural steel and dural affects the design.

THIN ALUMINUM ALLOY SECTIONS 49
S. A. KILPATRICK and O. J. SCHAEFER—In this first of a series, the authors show how to calculate compression members.

HOW TO MAKE ALIGNMENT CHARTS—II 53
M. G. VAN VOORHIS—Second part of series discusses use of logarithmic scales for various types of equations.

LARGEST AIRSHIP TO FLY IN FEBRUARY 56
First views of the structure and envelope of the LZ-129, the new German superliner of the air.

LUBRICATION OF SLEEVE BEARINGS—III 57
R. G. N. EVANS—Functions of mechanical lubricators, gravity circulating and pressure systems for centralized lubrication.

SIMPLIFIED HELICAL SPRING DESIGN 59
JOSEPH KAYE WOOD—A time saving method based on the load-deflection rate as a starting point.

LINCOLN-ZEPHYR BODY ASSEMBLED BY WELDING 61
Type of body construction made possible by recent developments in welding and welding equipment.

ALLOYS OF ALUMINUM AND THEIR USES 62
PAUL V. FARAGHER—A summary of the various types of aluminum alloys and their fields of application.

TYPICAL DESIGNS OF FLEXIBLE COUPLINGS—II 70
CYRIL DONALDSON—Flexible couplings wherein leather, metal disks or wire cable links provide the flexibility.

SHAPE FACTORS FOR ALUMINUM COMPRESSION MEMBERS 79
S. A. KILPATRICK and O. J. SCHAEFER—Shape factors for 10 commonly used sections for calculating their load carrying capacity.

Editorial 41
Product Developments 46
Question and Comment 67
News 72
Meetings 73
New Materials and Parts 74
Manufacturers' Publications 77
Books and Bulletins 78

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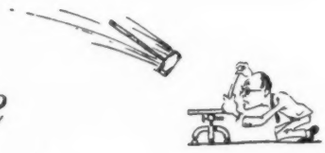
Intimate Correspondence



More Bouquets

PRODUCT ENGINEERING
NEW YORK

I have read with much interest the article of Cyril Donaldson entitled "Seven Rules for Good Appearance." I would suggest further such articles which might endeavor to explain the fitness of certain design elements for various applications, such as stationary, rotating, and moving objects. The articles might elaborate on surface texture and finishes and form a text on design principles for appearance which many designers might be glad to have. More bouquets for your excellent publication. —B.S.M.

New York, N. Y.

IT will interest B.S.M. and many other readers that we are now negotiating for a series of articles along the line that he suggests. Of course, we cannot guarantee that these articles will meet his specifications exactly, but we do know that they will be worthy of "more bouquets for your excellent publication."

Inaccurate Quick Reading

PRODUCT ENGINEERING
NEW YORK

Although the article "Bearing Metals for Severe Service" which appeared in the October and December numbers was commendable in some respects, like so many articles that have appeared elsewhere, they lose conviction because they are written to serve a selfish purpose and because of the circumscribed limits within which they present their facts. For example, in the article "Copper-Lead Bearings" which appeared in the October number, no mention is made of the difficulties encountered in casting copper-lead, nor does the article tell wherein copper-lead does not compare favorably with babbitt metal. —P.G.M.

New York, N. Y.

D. E. Anderson, author of the article referred to, is chief engineer of Bohn Aluminum & Brass Corporation. Before approaching him for the article in question, we were convinced that he was not unduly prejudiced with respect to copper-lead bearings in preference to the many other types of bearings that his company manufactures. We don't believe that P.G.M. read the article carefully. Mr. Anderson was quite clear in stating that it has been a "rather long and tedious process of developing a metallurgical and production technique that would insure consistent quality and a reasonable cost in large scale quantity." Quoting Mr. Anderson's article again, at the top of page 364, October *Product Engineering*, "a special method of pouring and a specialized cooling technique had to be developed. With this process the proper distribution of lead and copper is obtained." Of course, it was not the aim of the article solely to show wherein copper-lead compares unfavorably with babbitt and all other bearing metals.

Why Pick On 11 or 13?

PRODUCT ENGINEERING
NEW YORK

Table I on page 457 of the December number of *Product Engineering* gives "Characteristics of Three Bearing Alloys." This table compares Satco bearing alloys with SAE 13, lead-base babbitt and SAE 11 tin babbitt. Of course, Satco shows up favorably in this comparison which incidentally is made by C. H. Hack of the National Lead Company, which company, I understand manufactures Satco. A less prejudiced comparison would have been between Satco and a babbitt composed of 89 per cent tin, 7 per cent antimony and 4 per cent copper instead of SAE 11, and 10 tin, 15 antimony and 75 lead instead of SAE 13. —E.J.K.

Pittsburgh, Pa.

WE cast our votes with our contributor. Comparison with two standard alloys which are in wide use and the service performance of which are well established, gives a far more enlightening picture than a comparison with a "non-standard" alloy with which engineers in general are not so well acquainted and the differences are small.

For Intelligent Discussion

PRODUCT ENGINEERING
NEW YORK

Allow me to express my accord with your January editorial on a dead-line for society papers. The chief concern is that the papers should be secured in time to permit the issue of preprints. Without them, intelligent and widespread discussion of the paper is impossible. Certainly a society such as the A.S.M.E. does not need to beg and make concessions to obtain creditable papers on time! —E.S.A.

Cleveland, Ohio

The Path the Calf Made

PRODUCT ENGINEERING
NEW YORK

In the September 1934 number of *Product Engineering* on page 348 there appeared an anonymous poem entitled "The Path the Calf Made," which was contributed by one of your readers. This poem appealed to me so much that I would like to have it printed in our house organ. Of course, I am a subscriber and have a copy of the magazine in question. Therefore, if you will kindly give me your permission to have this poem printed in our company paper, I shall be indebted to your organization for another favor.

Also, I would very much like to know when my subscription expires as my records are not clear on this and I would not like to miss an issue of your magazine because of failure to renew through oversight. —J.T.

Los Angeles, Calif.

PERMISSION was granted. And we commend J.T.'s foresight in taking steps to make sure that he will not miss an issue through oversight and thereby break his file of back numbers.

Technical Data on Color

PRODUCT ENGINEERING
NEW YORK

In the article "Color Illusions" appearing in the January number, reference is made to Albert H. Munsell and his scientific formula relating to color. Will you kindly advise me if there is any scientific or technical data published by Mr. Munsell and if so, where it can be secured? —R.R.M.

Brooklyn, N. Y.

THE desired information can be procured from the Munsell Color Co., 10 E. Franklin St., Baltimore, Md.

Wanted—A Roll Top

PRODUCT ENGINEERING
NEW YORK

Could you tell us where we could procure a roll top constructed of steel or some other metal which might be used with a steel desk or cabinet. It is possible that we may want to utilize such a cover for new equipment which is being considered. If we find it ready made it will save our development department some work. —D.C.

Development Department

SOME questions we cannot answer. This is one. If any of our readers can help D.C. by telling him where he might get such a roll top, he will undoubtedly appreciate it.

Where Is the High Pressure?

PRODUCT ENGINEERING
NEW YORK

You certainly made a pretty looking presentation of the article by R. G. N. Evans on "Lubrication of Sleeve Bearings" but why does the author insist on going contrary to all accepted theory with reference to the position of the high pressure area? All authorities agree that it should be past the vertical center line instead of ahead of it.

In spite of this the article contains much valuable information on the lubrication of bearings. —G.A.A.

Baltimore, Md.

IF we were to edit the last sentence of the above letter we would say "All *other* authorities agree, etc." This, incidentally, brings up a question that will be featured in our March number as we have discovered that there is no complete agreement as to the position of the high-pressure area in sleeve bearings.

PRODUCT ENGINEERING

GEORGE F. NORDENHOLT, *Editor*

MCGRAW-HILL PUBLISHING CO., INC. FEBRUARY 1936 VOLUME SEVEN, NUMBER TWO

Professions and Unions Can't Mix

THE past depression offered many opportunities for unionizing. Some of the acts of the federal administration, directly or indirectly, encouraged labor organizers who naturally went after new fields to exploit. One of the latest was an attempt to organize the draftsmen and designers at the Chrysler motor plant. Although their efforts did not result in any evidence of success, the organizers did succeed in having a hearing before John F. Carmody, of the National Labor Relations Board. We were informed that the meeting was politely ignored by company and workers alike.

Repeated attempts have been made to unionize draftsmen, designers and engineers. In each instance the attempt has failed, not so much because of opposition from the employing companies but primarily because the men were not interested. Why?

Professional labor organizers simply cannot understand why these "white-collar" workers ignore the repeated efforts to organize them. One might even conclude that the union organizers gnash their teeth. Well they might when they contemplate the dues that might be collected, not to mention fines and perhaps a few "shake-downs" from the employers. It is a plum well-worth striving for and hope breathes eternal. Why give up the attempts to organize engineers and designers?

Several fundamental considerations point to the impossibility of unionizing draftsmen and designers. First, they are far above the level of the intelligence of the men with which labor organizers generally deal. Just as engineers analyze their professional problems, so do they analyze alluring promises, and the promises of the organizers lose their allure.

With few exceptions, the job of draftsman is but a stage in a career. It is not a static job but a step in the development to bigger undertakings. And these men see it in that light. You might just as well try to unionize hospital internes. Both engineering and medicine are professions, and in many respects the draftsman corresponds to the interne.

As for pay, of course we all want more. But effective unionization would mean so much for the job, regardless of ability. The man of exceptional ability would not receive his just rewards and he would be hampered by dictatorial rules. In short, creative work cannot be paid for equitably on a piece work basis. And it is only through creative work that the engineer can progress.

AS for working conditions and hours of work, it is hard to imagine how unionizing could improve them. Most manufacturing companies pride themselves, and rightly so, on the appearance, conveniences and congeniality of their engineering department. It is simply that practically all companies recognize that pleasant surroundings exert a strong influence on the character and quality of the work done.

Now, Mr. Labor Leader, please stop annoying these draftsmen and designers. Of course you have to make a living, but they also have their lives to live, their aims to achieve, their goals to reach. They have nothing in common with you. In short, they are not prospects for a trade union because they are apprentices in a profession where individual ability is rewarded.

Redesigning to use Aluminum

GEORGE W. DeBELL
Aeronautical Engineer
American Car and Foundry Company

Some of the outstanding points to be considered when designing with aluminum in order to attain the lightest structure possessing the required strength and rigidity. Structural steel and dural members of equal strength and stiffness are compared

PERHAPS the most common mistake made by designers in changing from one material to another is to fail to take into consideration all of the differences in the properties of the new material as compared with those of the material formerly used. There is often the tendency to consider only the difference in allowable unit stress and perhaps also the change in elastic modulus. But any one or more of innumerable other factors, such as heat-treating requirements, corrosion resistance, workability of the material, and similar characteristics may dictate certain design requirements which if ignored may cause early failure of the structure. When designing with the older materials, such as steel, the properties and propensities of which are well known, the engineer will almost instinctively take care of the many variables. However, when dealing with a new material one is likely to be so impressed with certain of its outstanding properties that other equally important properties are ignored. This is especially true when a structure is being redesigned to be made of a different material.

With the increasing importance being attached to light weight, many products made of steel are being redesigned to be made of aluminum. One of the developments of the past few years has been the design of truck and bus bodies, railroad cars and streamlined trains made of aluminum alloys. More and more engineers are facing the problem of redesigning their present products to be made partly or wholly of aluminum.

Whether or not it will be profitable to redesign equipment to be made of aluminum depends entirely on the relation of increased cost to increased value as measured by the purchaser and user of the product. In many instances increased value can be measured directly in dollars and cents. For example, if a railroad finds that because of decreased weight the savings in operating cost over the period of amortization are greater than the higher cost of the lighter equipment, the purchase of such equipment in preference to the heavier and cheaper equipment will be considered profitable. Similarly in airplanes, the value of light weight can be measured directly in terms of dollars by virtue of increased pay load and greater cruising radius. But it is not always possible to evaluate weight savings. Sometimes the weight saving adds to convenience, portability or other features not measurable in money. But in every instance the deciding factor is increased cost versus increased value as measured by the purchaser.

In spite of the fact that aluminum is only 1/2.8 the weight of steel, and even if cost comparisons are made on a unit volume basis rather than a pound basis, aluminum construction, based on current material prices, costs more than steel. Careful and exacting design analysis can reduce the cost of the aluminum structure

Fig. 1—Streamlined train on B.&O.R.R., built of aluminum by the American Car and Foundry Company. Note the unbroken surfaces at the inclosed vestibules

by savings in materials, and when it is considered that the primary reason for using aluminum is to save weight, it becomes all the more evident that the greatest care should be taken to reduce the amount of material to a minimum. This will require a careful investigation of possible buckling, local stresses, shear stresses, and similar factors not ordinarily considered in the design of steel structures.

Unfortunately, it has not been until recent years that great attention has been given to economizing in material. With steel, relatively cheap, a few added pounds were of no great consequence. Stress analyses were incomplete; "factors of safety" were high. If a member broke down it was simply made heavier, although the cause of its breakage might have been its excessive stiffness or the weakness of some of its associated member.

In most instances steel structures designed for strength are sufficiently rigid. Hence engineers whose designing experience has been confined primarily to the ferrous alloys have a tendency to disregard the effect of the stiffness factor. But when designing with aluminum alloys, dural, for example, the modulus of elasticity is only 10,000,000 as compared with 28,000,000 for steel. The ratio is closely 1 to 2.8. Thus, for equal unit stresses, aluminum will deflect about 2.8 times as much as steel. This simple fact, if not considered properly, may make the difference between a successfully engineered design and a regrettable failure.

Let us now compare structural steel members with aluminum alloy members having the same design, load and strength, bearing in mind the following basic relations:

1. The allowable stress, f, for the aluminum alloy dural and ordinary low-carbon structural steel is about the same.

2. The modulus of elasticity of structural steel E_S is 2.8 times the modulus of elasticity, E_A of aluminum alloy.

3. The specific weight of steel is 2.8 times the specific weight of aluminum alloy, dural. Steel weighs 490 lb. per cu.ft. and dural weighs 175 lb. per cu.ft.

4. The strength of members carrying direct stress, that is tension or short compression members, is proportional to the allowable stress f times the cross-sectional area A.

5. The stiffness of members carrying direct stress is proportional to the cross-sectional area A, times the modulus of elasticity E.

6. The strength of members carrying bending is proportional to the allowable stress f times the section modulus I/c.

7. The stiffness of members carrying bending is proportional to the modulus of elasticity E times the moment of inertia I.

In the following comparisons the subscript A is used to denote the aluminum alloy dural, and the subscript S is used to denote structural steel.

For direct stress members of equal strength,

$$f_A \times A_A = f_S \times A_S$$

But $$f_A = f_S$$

Therefore $$A_A = A_S$$

Since the areas in this case are equal the dural member will weight only 1/2.8, or 36 per cent, as much as the steel member.

For direct stress members having equal stiffness,

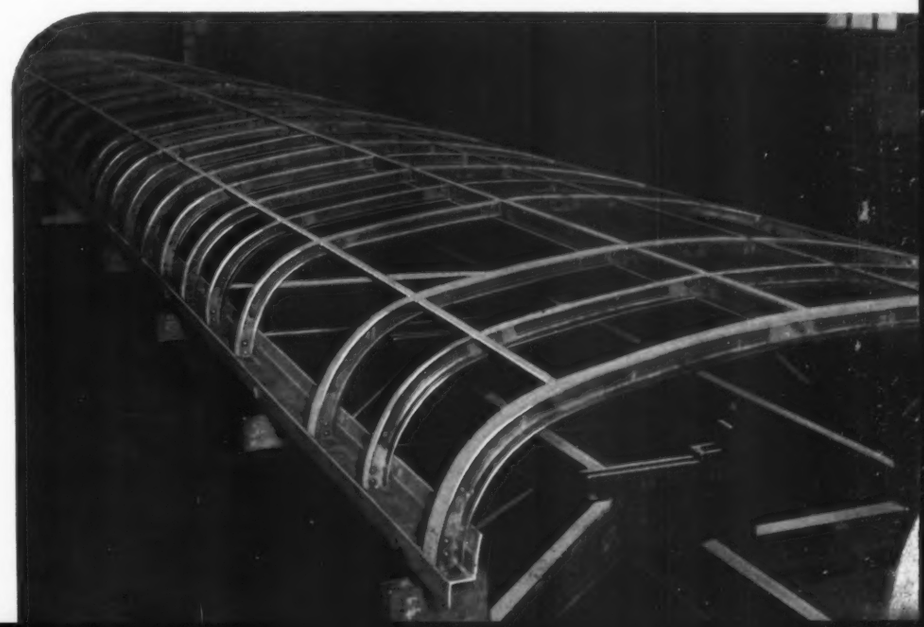
$$A_A E_A = A_S E_S$$

But $$E_S = 2.8 E_A$$

Therefore $$A_A = 2.8 A_S$$

It will be noticed that there is no weight saving by the use of dural, since the lower specific weight of the dural is exactly offset by the increased area required. However, it is unusual that the deflection of simple tension or short compression members be limited. In

Fig. 2—Aluminum framing for the roofs of the cars of B.&O. streamlined train built by American Car and Foundry Company



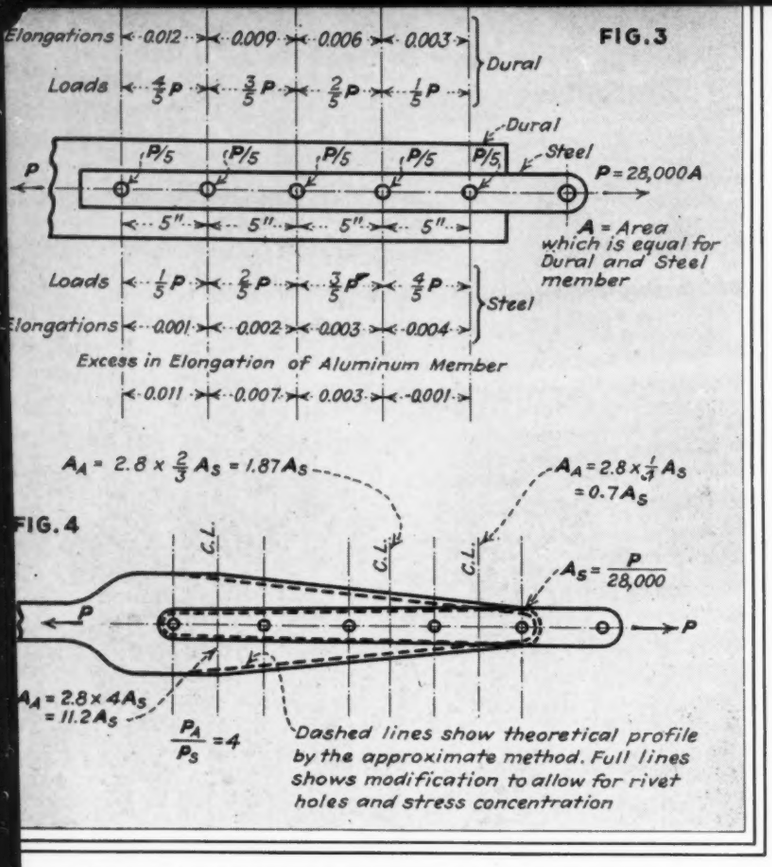


Fig. 3—Distribution of load among the various rivets will not be equal if sections of the connected dural and steel members are constant throughout their length

Fig. 4—An approximate method for equalizing the pitch elongation is dural and steel lap-jointed members

most instances such members can be designed without regard to stiffness, but where a stiffness requirement must be met the designer must adhere to the relation given above.

The comparison of members carrying bending is slightly more complex as the functions of section modulus, moment of inertia, and modulus of elasticity must be taken into account.

For bending members having equal strength,

$$f_A\left(\frac{I}{c}\right)_A = f_S\left(\frac{I}{c}\right)_S$$

But $$f_A = f_S$$

Therefore $$\left(\frac{I}{c}\right)_A = \left(\frac{I}{c}\right)_S$$

This indicates that the members are identical since their section moduli are equal and, therefore, the weight of the dural member will be as the ratio of specific weights. In other words, the dural member will weigh only 1/2.8, or 36 per cent as much as the steel member.

For bending members having equal stiffness,

$$E_A I_A = E_S I_S$$

But $$E_S = 2.8E_A$$

Therefore $$I_A = 2.8I_S$$

We must, therefore, for equal stiffness in bending use an aluminum alloy or dural member having a moment of inertia 2.8 times the moment of inertia of the steel member which it replaces. If the members are geometrically similar then, since the ratio of moments of inertia is proportional to the fourth power of the ratio of linear dimensions, the dimensions of the dural member will equal the fourth root of 2.8 times the corresponding dimensions of the steel member—that is, 1.295 times the dimension of the steel member. The area of the dural section will therefore be $(1.295)^2$, or 1.673, times that of the steel member. But since the specific weight of the dural is only 1/2.8 of the specific weight of steel, the dural member will weigh only 1.673/2.8, or approximately 60 per cent, as much as the steel member.

The use of geometrically similar members when converting from structural steel to aluminum alloy has the advantage that the form factors or section factors remain unchanged. But this type of conversion, of course, requires larger sized members in the aluminum alloy structure. In many cases the actual design requirements prevent the use of these larger sized sections with the result that a weight saving of 40 per cent is not generally obtainable since the substitution of shallower sections generally decreases the strength-to-weight and stiffness-to-weight ratios.

Let us now consider a theoretically perfect substitution and compare a structural steel member with a dural member having the same strength and stiffness in bending:

For equal strength, $$\frac{I_S}{c_S} = \frac{I_A}{c_A}, \quad \text{since } f_A = f_S$$

For equal stiffness, $$I_A = 2.8I_S, \quad \text{since } E_S = 2.8E_A$$

Hence, $$\frac{I_S}{c_S} = \frac{2.8I_S}{c_A}, \quad \text{or } c_A = 2.8c_S$$

In order to illustrate the effect of this requirement, as shown in the above equation, let us consider two rectangular bars, one of structural steel and one of dural, both having the same strength and stiffness. Since the I/c of a rectangular bar equals $bd^2/6$ and $d = 2C$, we can write that,

$$b_A c_A^2 = b_S c_S^2$$

But $$c_A = 2.8c_S$$

Therefore $$b_A(2.8c_S)^2 = b_S(c_S)^2$$

$$b_A = \frac{b_S}{(2.8)^2} = \frac{b_S}{7.84}$$

This means that the dural bar, for equal strength and stiffness, would have to be 2.8 times as deep as the steel member and only 1/7.84 as thick. The weight would be in the ratio of about 1 to 7.84 in favor of the aluminum section. The above, however, is a purely theoretical deduction. In almost every instance it would be impossible to reduce the thickness of the section to 1/7.84 of the thickness of the steel section, especially with the depth of the section multiplied by 2.8. Failure would most likely occur through local buckling or the section might twist on account of lack of stability, since the stiffness or stability of a flat surface is proportional to the square of the thickness and to the modulus of elasticity. Finally, it will usually be found that there is insufficient room to increase the depth of the beam 2.8 times.

On the other extreme, the futility of substituting an aluminum beam of the same depth and stiffness as the steel beam is readily illustrated by a specific example. Considering an I-section and neglecting the moment of

inertia of the web, the moment of inertia of the section would have to be increased 2.8 times by increasing the size of the top and bottom flanges. If the radius of gyration of the flange is to remain the same, it will be necessary to increase the area of the flanges 2.8 times and the increased area would have to be obtained by increasing the width of the flanges. This would rarely be possible, because it would result in such a wide flange that the free edges would buckle. Therefore, it would be necessary to secure the added area by increasing the thickness of the flange as well as increasing its width. This would result in a decrease in the radius of gyration of the flange section, thereby requiring an additional increased area. The equivalent section would then have more than three times the area of the steel section and there would be no saving in weight.

The actual substitute decided upon in engineering practice is usually a compromise between the theoretically perfect substitute and the wholly practical one. Since is is necessary to compromise it is important that the stress analysis be complete so that consideration can be given to all possible effects and full advantage taken of weight saving possibilities.

Still further complications arise when aluminum and steel members are used in combination over an appreciable length of the structure. When an aluminum and a steel member are connected by a lapped joint that extends for an appreciable length, it is absolutely essential that the differences in the deflection rate of the two materials be compensated.

The accompanying Fig. 3 shows how a steel bar might be connected or riveted to an aluminum member. For purposes of illustration, five rivets are shown. It is also assumed that the load is tension and that the cross-sectional areas of the aluminum member and the steel member are the same. If the rivets are to share the load equally, each will have to carry a load of $P/5$. The load distribution at the various pitches in the steel bar and aluminum plate are indicated in the figure.

Taking the first panel pitch, the load in the dural member is 4 times the load in the steel member. If it be assumed that the load P stresses the material to its maximum, the unit elongation for the steel under the maximum load will be 28,000/28,000,000, or 0.001 in. per in. This will be for a load of P. Similarly, the aluminum member will elongate approximately 0.003 in. per in. of length. For 5 in. lengths, these elongations will be 0.005 and 0.015 for the steel and aluminum, respectively.

As can be seen from the dimensions given in the figure, the difference between the pitch distance in the steel member and the corresponding pitch in the dural member is 0.011 in., 0.007 in., 0.003 in. and —0.001 in. Evidently, under these conditions, the load cannot be distributed equally among the five rivets. And in any event, because of the unequal deflection of the steel and aluminum members under load, the rivet holes will be thrown out of alignment. Although this will amount to only a few thousandths of an inch, up to a maximum of 0.011 in., under repeated loads it will cause the rivet holes to elongate. In the example given, this action will begin with the first rivet to the left. Eventually, the loose rivet will tend to transfer its load to the next rivet, causing this rivet to loosen in turn. In this manner, the successive rivets will loosen and fail.

To overcome this condition, the steel and the aluminum members should be tapered to give equal deflections, as indicated in Fig. 4. Theoretically, in each pitch the area of the aluminum section can be calculated by the following formula:

$$\frac{P_A}{A_A E_A} = \frac{P_S}{A_S E_S}$$

$$A_A = \frac{E_S}{E_A} \times \frac{P_A}{P_S} \times A_S = 2.8 \frac{P_A}{P_S} \times A_S$$

In this equation P is the load transferred over the pitch distance, the section of which is being calculated. The area of the cross-section of the steel member at and beyond the last rivet to the right will be required to carry all the load and its cross-sectional area will be:

$$A_s = \frac{P_s}{28{,}000}$$

To the requirement of equal deflection, the area of the cross-section of the aluminum alloy member at the last rivet to the left must be determined from the deflection equation and not from a stress standpoint. Acually the stress in the aluminum alloy member at this point will be much lower than its allowable stress in order to satisfy this deflection requirement. In order to have both the steel and dural members working to full capacity it is necessary to use a heat-treated steel whose allowable stress is 2.8 times the allowable stress in the dural member; in other words, the ratio of stresses must equal the ratio of moduli of elasticity. It is a relatively simple matter to calculate the required areas in the various pitches from the above equation, but it should be remembered that these areas are the average, or mid-point, areas for each of the pitches and not the areas at the rivets. It is, therefore, necessary to check the net area at each rivet to determine if it has sufficient strength to pass the required load without exceeding the allowable unit stress. In the majority of cases a straight taper will give a sufficiently close approximation of the required area.

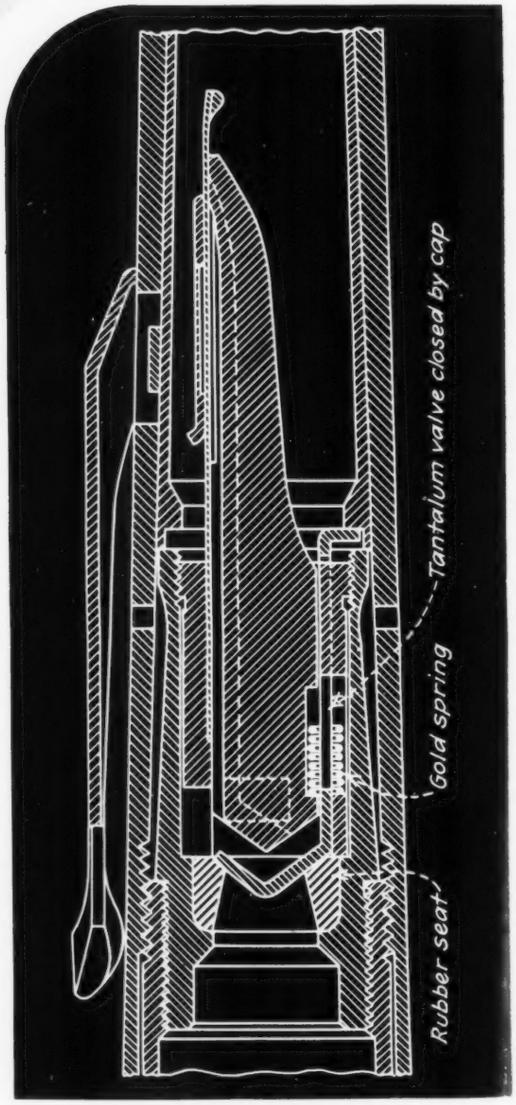
Fig. 5—Interior of baggage car of B.&O. aluminum streamlined train. The sheathing is corrugated to prevent denting by baggage

PRODUCT DEVELOPMENTS

Tantalum Resists Corrosion in Fountain Pen Valve

When the cap is placed on the Wahl fountain pen, ink is sealed within the barrel by a tantalum valve

Tantalum for the valve resists corrosion from the ink and from contact with rubber compounds



A stamped tantalum valve seals the ink in the new Wahl pen when a shoulder on the cap is screwed down against the small extension of the valve

● Relatively small change in temperature of the air within fountain pen barrels has always made it difficult to prevent leakage of ink. When the pen is in such a position that the passage to the point is submerged, expansion of the air within the barrel forces the ink out. But to seal the ink into the barrel when the pen is not in use is a difficult problem because of the tight seal required and the corrosive action of the ink.

To seal the ink in the barrel, a metal valve is used in a new pen developed by engineers of the Wahl Company. The necessary sealing pressure is obtained by engaging a projection on the valve against a shoulder on the cap as shown by the accompanying diagram. As the cap is screwed tight, the mechanical advantage of the screw results in a relatively high pressure against the valve seat. When the cap is removed, a small gold spring opens the valve and allows ink to flow to the pen.

Corrosion of the metal valve is caused not only by the ink but also from contact with two different types of vulcanized rubber. The valve seat is made of semi-soft rubber, while the cap which bears against the other end of the valve is made of hard rubber. The best stainless steel rapidly becomes pitted and corroded in contact with the rubber composition. Solid gold successfully resists corrosion but it is too expensive at present market prices.

The material finally selected for the valve is tantalum, a heavy metal having physical properties comparable to steel but characterized by high resistance to acids and chemicals present in ink and rubber compounds. Valves are stamped from rolled sheet and further hardened by heat treatment, giving physical properties equal to parts made from cold rolled steel. Thus, the tantalum valve is not only stronger and harder than solid gold, but it resists corrosion as well and also costs much less. With high seating pressure, leakage is eliminated.

• *Steel Mill Bearings*

In a 4-high finishing stand of a new 80 in. mill, backing roll bearings made by the Bantam Ball Bearing Company have a load capacity of 3,400,000 lb. each at 100 r.p.m. These 30-in. wide bearings are said to be operating under the highest load per unit of width on any existing hot strip mill.

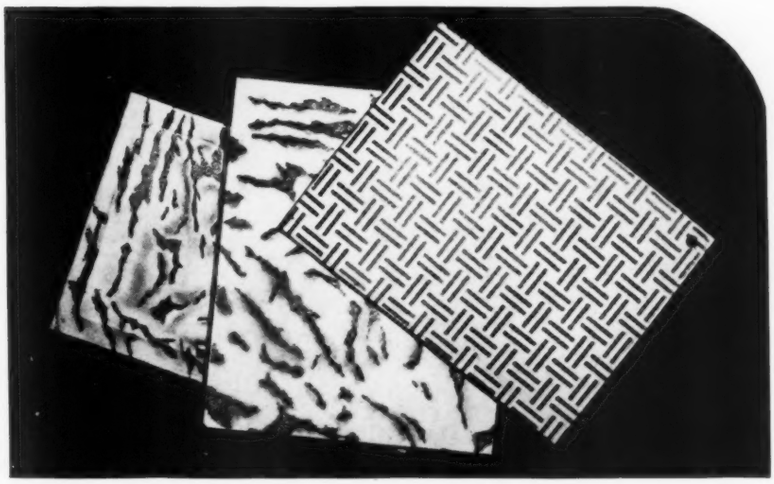
• *New Enamel Finish*

By applying sprayed molten metal to unfused porcelain enamel sheets, new decorative effects in porcelain enamel finishes are obtained in a process developed by I. V. Brumbaugh. The sheets are prepared in the usual way with a ground coat that may or may not be fused. Then, molten metal is sprayed on the sheet using a stencil if patterns rather than solid color effects are desired. When fused, the sprayed metal combines with the enamel to produce color effects shown in the accompanying illustration.

For the piece having a wicker work design, copper was sprayed on unfused ivory colored enamel. Then, an overglaze of clear glass was applied and all fused together. For the marble effect, stainless iron was sprayed on unfused white, and then covered with an overglaze and fired. The blending between the stencil openings resulted from iron dust blown under the stencil by the blast from the gun.

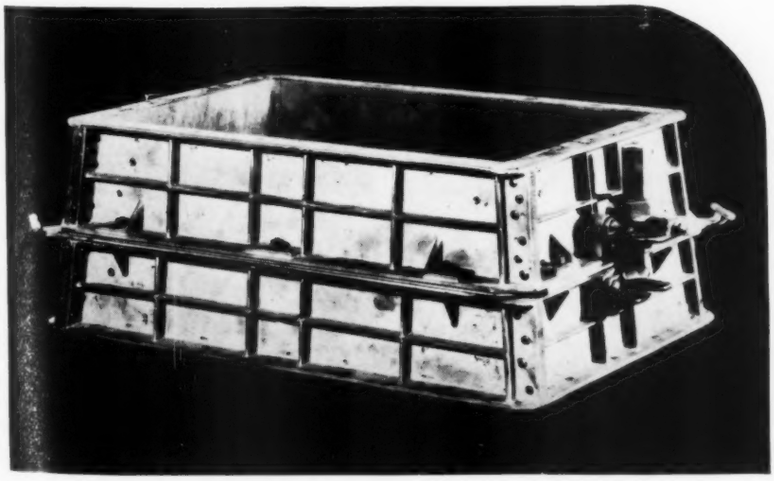
For the border effect obtained in the third plate, the outer rim of white was fused first. Then the center was filled with unfused white and iron sprayed over the whole sheet. No overglaze was used.

Only a small amount of metal is required. Because of the action of the flame in the spray gun, nearly all of the metal is oxidized, and these oxides are readily incorporated into the enamel. Various color effects thus can be obtained from different metals.



Decorative effects obtained on porcelain enamel by applying sprayed metal before final firing

Riveted corner design and ribbed sides stiffen magnesium alloy foundry flasks



• *Magnesium Alloy Molding Flasks*

Because foundry flasks have to be lifted and carried by hand, light weight is desirable. But, the sides of the flask must also be strong enough to withstand bending stresses set up in ramming the mold. To meet these requirements, a new flask designed by the American Foundry Equipment Company, is fabricated from Dowmetal with a design incorporating a strengthened corner and stiffened side wall.

As compared with an aluminum flask of the same dimensions, the magnesium alloy flask weighs 30 per cent less. The actual weight of a flask 12 by 18 in., 10 in. high, is 30 lb. including both cope and drag. The riveted corner construction and the ribbed side wall design, shown in the accompanying illustration, gives added strength and stiffness.

• *Light-Weight Riveter*

In designing a portable hydraulic press for driving and squeezing rivets up to $\frac{3}{8}$ in. in diameter, aluminum was extensively used by the Hanna Engineering Works to improve the portability of the unit.

Although an aluminum alloy yoke would have resulted in a further decrease in weight, steel was selected for this part for compactness. The press was designed for use on such jobs as riveting automobile frames. Limitations in the size of the yoke required the use of steel for this part since the section of an aluminum yoke of the desired stiffness would be too large.

• *Portable Placer Drill Travels by Air*

For prospecting in inaccessible places, airplanes are being extensively used today, and prospecting equipment therefore must be designed for air transportation. This limits both the length and the weight of individual pieces from which a

portable drill rig can be assembled.

In the placer drill shown here, designed by the C. Kirk Hillman Company, no piece weighs more than 150 lb., nor is longer than 6½ ft. The drill weighs only 2,000 lb., complete with tools and casing for drilling a 5¼ in. hole 200 ft. deep.

The gasoline engine drive was simplified by combining clutch and speed reduction in a multiple V-belt drive. The engine is pivoted at the rear and can be raised by a foot pedal to disengage the V-belt from the aluminum driving pulley. This simple drive arrangement also eliminates shock caused by lifting and dropping the tools and makes adjustments unnecessary. Not only is weight and space saved, but the only spare part needed for the drive is one set of belts.

• *Random Jottings*

In centrifugals made by Chemicolloid Laboratories, Inc., a protective layer of Stellite is deposited on the mill rotor and on the nickel steel driving shaft. The hard facing not only reduces shaft wear at the gland but also resists corrosive attack from materials being processed.

• • •

West Wind ventilating fans have all steel parts Udylite-Cadmium plated for appearance and to resist corrosion. Some are brushed to a satin finish for final finish, and others are color lacquered over the cadmium.

• • •

Compressed air is used to actuate 24 hold-downs that clamp work under a total pressure of 120 tons in a mammoth plate planer recently built by Baldwin-Southwark Corporation. For the first time on a plate planer carriage reversal and feed are electrically controlled.

• • •

Hot liquids, acids and alkalis encountered in sugar refining processes quickly break down ordinary paint films. A new primer and finishing coating made from bakelite resin has demonstrated that these finishes have marked superiority in resistance to corrosion.

• • •

In several Rannett pumps now being installed on a gasoline distribution line between Des Moines and the Twin Cities, the electric motor is cooled by the gasoline pumped. By proper sections and design, nickel cast iron cases successfully withstand pressures up to 1,500 lb. per sq.in. without leaking.

In a demonstration device built by Synthane, gear vibration is made visible by the use of a microphone and a cathode ray oscillograph. Noises are reproduced on the screen of the tube giving a more accurate comparison than can be obtained by ear.

• • •

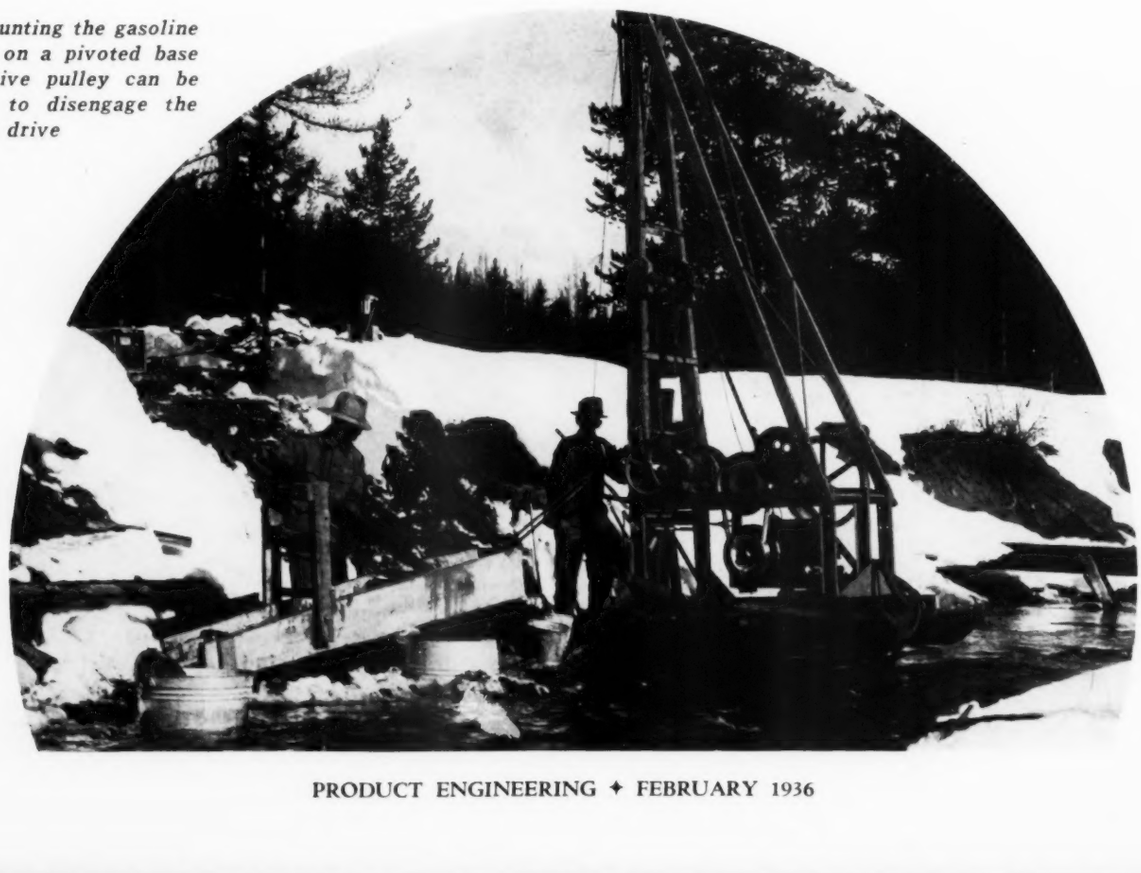
Rotors of meters used to measure boiler feed water are subjected to highly erosive and corrosive conditions since they are exposed to the action of hot, dirty, blow-down water from boilers. The Henszey De-Concentrator Company is now using Inconel as standard equipment for meter rotors, not only for feed water meters but also for devices measuring chemicals.

• • •

Carrying capacity of skips recently installed at the International Nickel Company's Frood Mines was increased by more than 60 per cent by replacing carbon steel with nickel steel. The capacity of each load was increased by 1½ tons with no change in the size of hoisting cable nor in the amount of power consumed.

• • •

To eliminate "saddened" shades which sometimes result from metallic action on dyestuffs, the Franklin Process Company has equipped all its plants with stainless-metal package-dyeing machines.



By mounting the gasoline motor on a pivoted base the drive pulley can be raised to disengage the V-belt drive

Stress Calculations for Thin Aluminum Alloy Sections

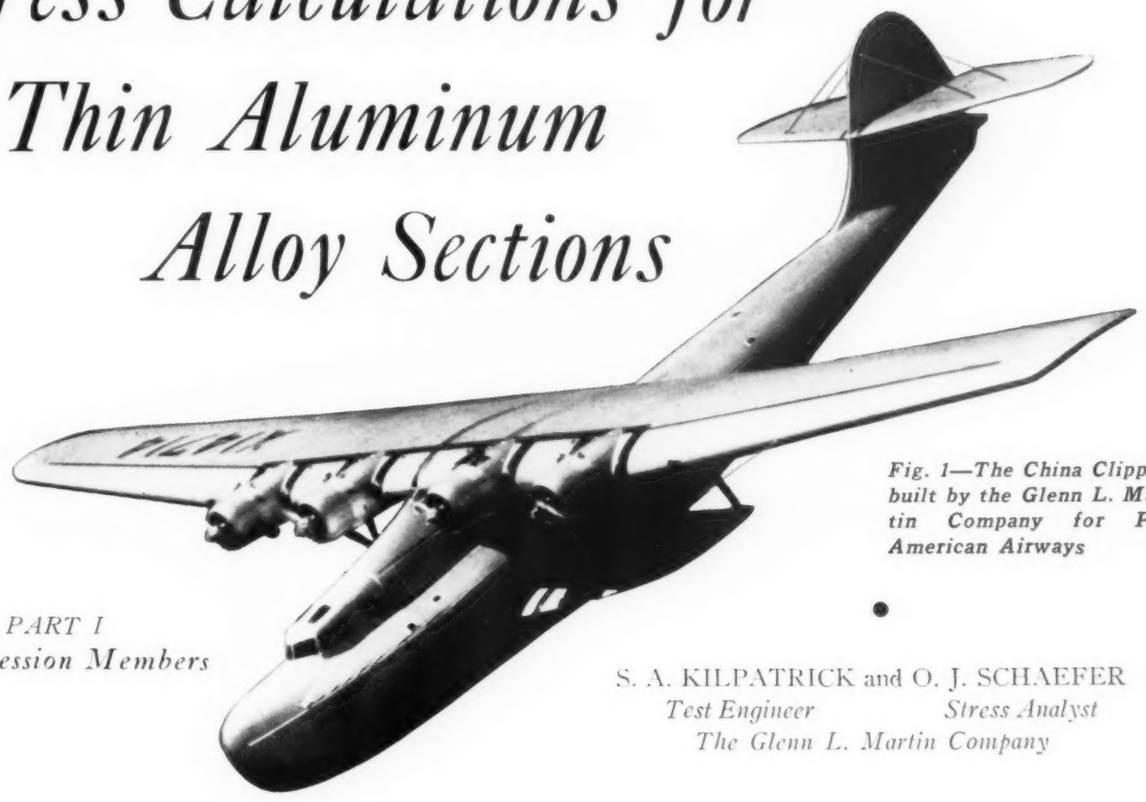


Fig. 1—The China Clipper, built by the Glenn L. Martin Company for Pan American Airways

PART I
Compression Members

S. A. KILPATRICK and O. J. SCHAEFER
Test Engineer *Stress Analyst*
The Glenn L. Martin Company

This series of articles deals with the fundamentals of sheet metal structural design as applied to aircraft and summarizes the latest developments. Some of the information presented has been obtained by the cooperative research of various organizations, but much of the data resulted from analytical studies and test research conducted by the Glenn L. Martin Company. Methods and formulas given here have been checked by full-size static test specimens. —B. C. Boulton, *Chief Engineer The Glenn L. Martin Company*

IT IS OF utmost importance for a designer to be able to tell quickly the relative efficiency of various sections with respect to their strength as compression members. With this in mind, the method described here was developed so that with certain basic test data available, any section can be evaluated and indexed for ready reference. This method makes it possible to predetermine without further tests, how a section of any length or gage thickness will behave.

Compression members are far from simple. When the member is short it must be designed efficiently to resist local buckling. When it is long it must have sufficient size to resist an elastic column failure. The transition from failure as a short column to failure as a long column is not abrupt. In a certain length range failure takes place both by local buckling and elastic bending. Compression members are always a compromise when length is a factor. An efficient design in long length is usually least efficient for short lengths.

For any section, it must be known how the column will behave under load for all ranges of column lengths as measured by the ratio of length to radius of gyration. For example, if we study the section shown in Fig. 3 from the standpoint of failure we must know immediately how it is employed and the kind of restraint afforded. A section of this nature would generally be used as a beam chord (or possibly a stiffener) such that a web is always present between the flanges marked *a*. If this section be tested as a long column with a ratio of length to radius of gyration of more than about 100, the failure will be purely elastic. The column will bend about the *y-y* axis. Tests on successively shorter columns will show that as the length decreases the legs *a* will begin to buckle elastically, although the actual failure may still be in bending.

With further decrease in the length of the column, the flange *b* will bow in the direction indicated by the arrows, and flange *a* will buckle, thus causing failure. At the same time, the first type of failure will still be noticeable. The significance of this is that although the flange *a* had failed potentially at a relatively low load, it continued to remain quite straight until flange *b* had failed elastically about an axis at 90 deg. to that about which *a* had failed. Such a section has in effect two degrees elastic freedom, one being more predominant in the short range and the other in the long range.

It follows that every type of section may possibly have a different load curve in the short column range. For certain comparatively simple sections theoretical solutions have been obtained, but for the more complex

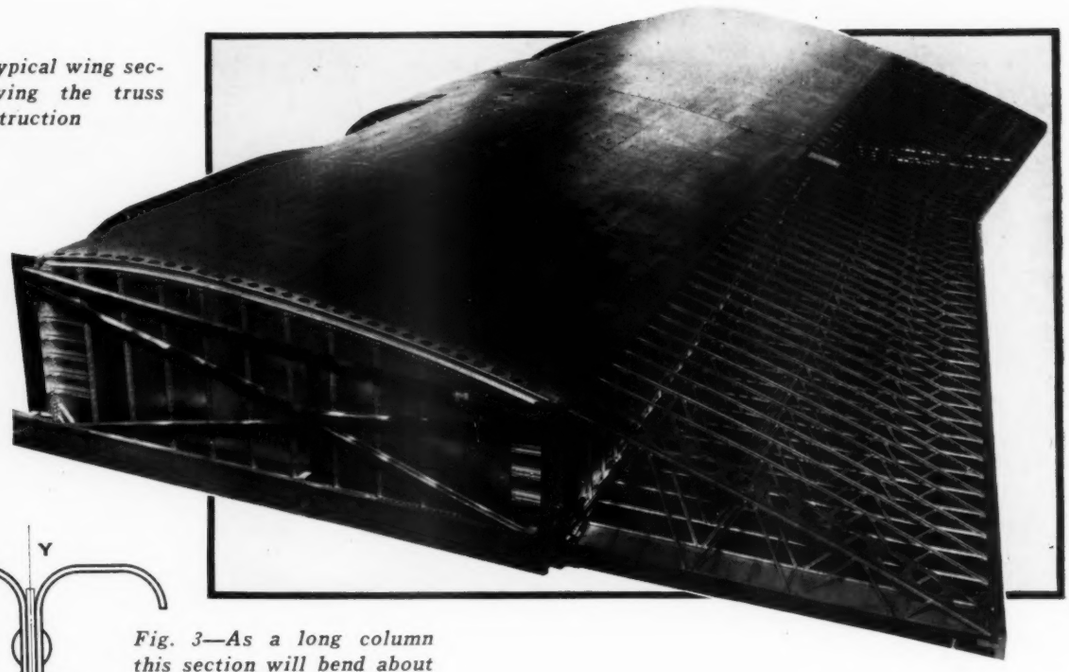


Fig. 2—Typical wing section showing the truss type construction

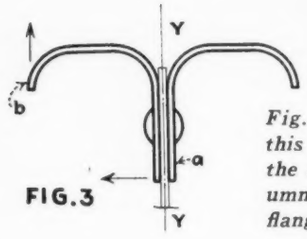


FIG.3

Fig. 3—As a long column this section will bend about the axis y-y. As a short column, flange b will bow and flange a will buckle

sections we must still rely on empirical methods or tests.

In plotting the results of tests on certain sections for various column lengths and gages of sheet, there will be obtained a series of curves similar to those shown in Fig. 4. In these curves the stress at failure, P/A, is plotted against the thickness t of the sheet for given values of the ratio of length to radius of gyration, L/ρ. Many formed sections will have curves of similar shape. In each instance the family of curves can be represented empirically by the equation of the hyperbolic tangent such that for any given value of L/ρ there will be the relation:

$$P/A = \sigma \tanh (Kt) \qquad (1)$$

In this equation

σ = limiting stress for each curve of Fig. 4
tanh = hyperbolic tangent
K = constant determined by experiment
t = thickness of the material in in.

There are some sections, such as simple angles and open channels, or others where the external dimensions are large in proportion to the gage of the sheet, for which the stress curve can be closely represented by the equation,

$$P/A = \sigma \tanh (Kt^2) \qquad (2)$$

In this equation the curves will take the form shown in Fig. 5. For sections represented by this equation the limiting values of σ may appear higher in a particular instance than for some other sections whose allowable stresses are calculated by Equation (1). The explanation for this will be given in a subsequent paragraph.

If the constant, K, and the limiting stress, σ, are known, the above equations enable the designer to quickly determine the failing stress P/A. This constant K, called the shape factor, has been determined for numerous shapes of sections. Its determination requires merely that the given section be tested as a short column with the value of L/ρ from 15 to 20. The chart shown on pp. 79 and 80 gives the shape factors for nine commonly used sections. The value of σ can be calculated by use of a suitable column formula whose limits are the ultimate compressive stress of the material at zero length and the Euler curve at the longer lengths. Many equations have been used for the short column range, but since no reliable method is known for making a proper selection, two equations are presented here which will give about the maximum and minimum limits for all sections.

Tests on sections similar to Fig. 3 have shown that for the normal range of gages the equation for σ can be expressed by the Rankine formula which is

$$\sigma = \frac{f}{1 + B} \qquad (3)$$

where σ = allowable stress in lb. per sq.in.
f = ultimate compressive stress of material, generally taken as the yield point.

$$B = \frac{f\,(L/\rho)^2}{C\,\pi^2\,E}$$

L = length
ρ = radius of gyration
C = coefficient of end restraint

Equation (3) would be somewhat conservative if flange a of Fig. 3 is restrained by closely spaced stiffeners or other suitable means. For very compact sections, tubing and corrugated sheet and for the very simple sections, a higher limiting curve for σ may be used. The equation recommended is a modified form of the Rankine formula as derived by Natalis:

$$\sigma = \frac{f\,(1 + B)}{1 + B + B^2} \qquad (4)$$

It can probably be said that, in general, where a section does not show false limits, as will be explained further on, Equation (4) is the more correct.

In working out the results of tests on two sections of widely different shapes and proportions, one with flanged angle sections and the other with corrugations,

it was found that the value of K varies with the ratio of L/ρ. This is logical. The basic or limiting curve for a given section takes the form of Equation (3) as indicated by the upper curve in Fig. 6(a). The lower curves are the curves of allowable stress at given values of thickness. Each of these curves approaches the basic curve as a limit when L/ρ is increased until, at high values of L/ρ only the lowest gages are below this limiting curve. If then, at a large value of L/ρ, the stress P/A is plotted against t, as in Fig. 6(b), the value of K in Equation (1) must be relatively large. An equation which represents quite closely this variation for the two special cases of flanged angle section and corrugation is:

$$K = K_o\left(\frac{f}{\sigma}\right)^{\frac{1}{2}} \qquad (5)$$

K = constant as determined from Equation (1) or (2) for any give L/ρ

K_o = constant for short lengths at about $L/\rho = 20$

In order to understand the procedure of handling these equations, an example will be shown. Assume a section similar to Section 2 in the table of shape factors (pp. 79 and 80) having a sheet thickness of 0.051 in., and L/ρ of 50. The section is assumed to be employed where straight edges are restrained.

$K_o = 12$, $f = 50{,}000$ (actual), $L/\rho = 50$ $E = 10{,}500{,}000$

The basic curve is chosen as Equation (1) with $C = 1$. Then,

$$\sigma = \frac{50{,}000}{1 + \dfrac{50{,}000}{\pi^2 E}(50)^2} = 22{,}700 \text{ lb. per sq.in.} \qquad (6)$$

From Equation (5),

$$K = 12\left(\frac{50{,}000}{22{,}700}\right)^{\frac{1}{2}} = 17.8$$

Applying Equation (1)

$$P/A = 22{,}700 \tanh (17.8 \times 0.051)$$
$$= 22{,}700 \times 0.72 = 16{,}200 \text{ lb. per sq.in.}$$

Not only does this method give reasonable values, but relatively small errors will result even with a relatively large error in the value of K selected. For instance, had a value of 15 been chosen for K_o, in the above example, instead of 12, the allowable stress would have calculated about 18,500 lb. per sq.in. Thus, an error of 25 per cent in the selection of the value of K_o would result in an error of only 14 per cent in the allowable stress.

The shape factors shown in the table (pp. 79 and 80) are for use with Equation (1). The test yield points are also given in the table for comparative purposes, but for design purposes the guaranteed value should be used except where definitely known. In this connection, however, it should be noted that in many compact formed sheet sections, the true yield point will be higher than that for the sheet stock, because the cold working of the metal increases the yield point.

No complete families of complex sections were available for the evaluation of K in terms of dimensions. The calculations were made for only simple angles and flat sheets. For angles, the equation for K is quite complex. The theoretical equation for allowable stress, as shown by E. E. Lundquist in National Advisory Committee for Aeronautics, Technical Note No. 413, is simple to use, but the empirical values are given here because they express finite limits at all length, including zero length.

In the tests on angles in the short range, it was found primarily that the basic curve could not be expressed by either Equation (3) or Equation (4) nor by the familiar Johnson parabolic. It was, nevertheless, a faired curve becoming tangent to the Euler curve (with C slightly greater than 1) at large values of L/ρ. Other values of σ between L/ρ = zero, and the Euler curve are given in the following tabulation:

L/ρ	σ	L/ρ	σ
0	40,000	60	27,000
20	38,000	80	18,000
40	34,000		

The equation for the allowable stress for such angles was then

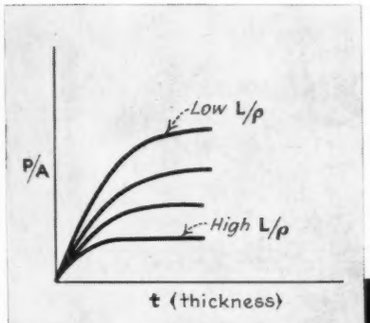


FIG.4

Fig. 4 — Relation between stress at failure, P/A, and sheet thickness t. Fig. 5 — Stress curves for sections where external dimensions are large in proportion to the sheet thickness

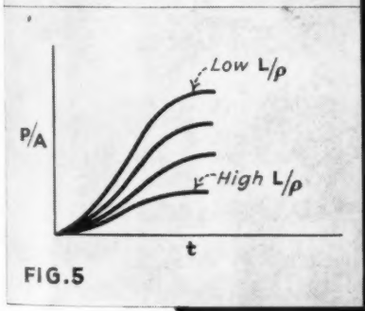


FIG.5

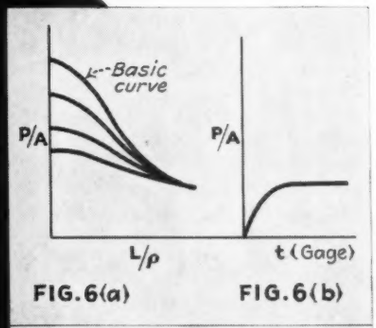


FIG.6(a) FIG.6(b)

Fig. 6—Upper curve in (a) represents Equation (3). Lower curves are for increasing L/ρ. In (b) is the curve for large L/ρ with P/A plotted against t. Fig. 7—Curves of allowable stress for the section shown in Fig. 3 showing the effect of sheet thickness

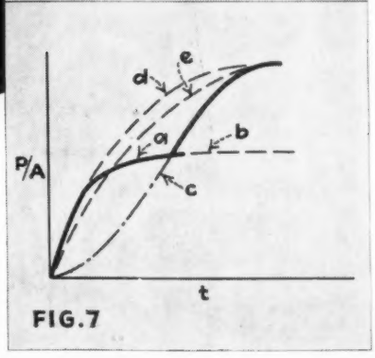


FIG.7

found to be given by the following relation:

$$P/A = \sigma \tanh K \left(\frac{t}{b}\right)^2$$

In this equation b is the width of the leg. The value of K can be expressed in terms of length and radius of gyration such that,

$$K = 149.1 + 0.1\left(\frac{L}{\rho} - 47\right)^2$$

This equation only holds for $C = 1$, test data not being avialable for greater end fixity.

It has been indicated that the limiting stress σ for certain sections whose allowable stresses are calculated by Equation (2) may appear higher than for some section as calculated by Equation (1) when L/ρ is held constant. It was also pointed out that false limits could be obtained for certain types of sections. To gain a clearer picture of these points the section of Fig. 3 will be used as an example, confining the discussion to only short lengths for the present. In practical design, sections of similar external dimensions are generally employed. That is to say, the dimensions are not varied in direct proportion to changes of thickness. The section of Fig. 3 was improved considerably by turning a small flange along the edges. For small gages and short lengths this flange acts as an excellent reinforcement.

If the gage of the section shown in Fig. 3 is increased without increasing any of the other dimensions it can be seen that a point is reached where the flange along the edges would do little good, since the flange width is approaching the thickness of the sheet as a limit. The curve for allowable stress for this condition is represented by curve *a* of Fig. 7. This particular section would therefore show a high shape factor but a low limit as represented by *b* in the figure. For the higher gages the allowable stresses would follow curve *c*, thus producing a composite curve for the whole range. Curve *c* will be recognized as being similar to the curve for a plain angle without flanges. If the reinforcing flange had been gradually increased as a function of the thickness curve *d* would have been the result.

For certain sections the normal range of gages used would be entirely included by the lower curve in Fig. 7, whose limit may fall below the ultimate compressive stress and even below the yield point. Here, then, in order to use Equation (1), σ would necessarily be taken as the lower limit. It will be seen, therefore, that simple sections such as the plain angle will have only one limit for the whole range of gages. As a consequence the limits for a section generally regarded as very inefficient will be higher than for some sections considered quite efficient.

In the foregoing example a constant length was assumed. With each increase in length the curves will be similar to those of Fig. 7, but the difference in limits will become decreasingly pronounced until both limits finally merge into the Euler column curve.

Of the sections selected for the table of shape factors (pp. 79 and 80), only Section No. 5 showed a pronounced low limit where the $\frac{1}{4}$-in. flange was used. To obtain a faired curve for all gages it is recommended that the flange be increased with the gage from about $14t$ at zero point on the thickness ordinate to $4.5t$ when the thickness of sheet is 0.090 inch.

As a generality, if a section of high shape factor, such as Section No. 4 in the table, is doubled in external dimension, it is reasonable to assume that the shape factor will be reduced. The question of how much this reduction amounts to cannot be determined exactly without test, but in this instance, it would seem logical to use half the value, or $K_a = 13.5$. When a section such as No. 1 in the table is doubled in external dimensions, however, halving the value would make $K_a = 7.8$. Any value below 10 should be viewed with suspicion. In this instance, it would be reasonable to suppose that the section no longer followed the law of Equation (1), but perhaps followed Equation (2). A reasonable value for K_a would then be $(15.6/2)^2$ or about 61, to be used with t^2 in place of t. The limiting value for σ would then possibly rise above that used for the section with the higher shape factor.

A section similar to No. 3 in the table would be very poor without ample fixity along one edge as represented by the wood block in the tests. Close stiffener spacing would approximate the condition when it is used as a beam chord. Without this, the equation for simple angles would be the more nearly correct.

It has been stated with relation to sections that the most efficient ones are those with the least amount of flat area. Sections of continuous curvature are the ideal but are not always the simplest to use. For this reason, for highest efficiency a section should be designed with the amount of flat area only great enough to allow simple attachments to other members. Where members are used as columns and large values of radius of gyration are required it is sometimes necessary to use fluting or lightening holes along the flat sides.

Corrugated sheet is as nearly ideal as a section can be made. The data in the table (p. 79 and 80) is only shown as a comparative check. More complete information on this section will be found in the references. It was shown by E. E. Blount (Ref. No. 1) that the optimum ratio of pitch to depth is 3.13 for short columns and 2 for long columns.

The shape factors listed in the table are not to be understood as applying to all materials. The effect of physical characteristics was not investigated, but K will be a function of E as well as the yield point.

[Further information on various sections can be found in Air Corps Technical Report No. 3739. These sections were not evaluated for K because of insufficient data.]

To be continued in the March number

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How to Make Alignment Charts

M. G. VAN VOORHIS

Construction of charts for solving the equations $xy=z$, $x/y=z$ and $xy=uv$ graphically, and the use of logarithmic scales. In the March number Z charts will be discussed

PART II

GRAPHIC solutions of equations containing multiplication and division of variables and functions of variables are most frequently desired. The construction methods which follow in Figs. 6 and 7 are similar in phrasing to those which have been given in Fig. 1. A comparison shows that relatively few words have been changed, yet logarithmic scales are used in the following nomograms whereas uniformly divided scales were used in Fig. 1 nomogram.

Nomogram for $xy=z$ (Fig. 6)

Draw two parallel lines. Subdivide one line with a logarithmic scale for x within the limits given and the other line similarly for y, the two scales ascending in the same direction. The size of each scale is independent of the other and can be as large as the space allows for the given limits. The z scale will always be on a line parallel with and between the x and y scale lines under the above conditions of layout.

CONSTRUCTION METHOD FOR LOCATING THE z LINE—Select a value of z within the range of its scale. For each of the two selected values of x, determine the value of y to satisfy the equation with the selected value of z. Draw two lines connecting the corresponding determined values of x and y. The point of intersection locates the z scale line and will be the selected value of z on the z scale.

FORMULA METHOD FOR DETERMINING DISTANCES BETWEEN SCALE LINES IN TERMS OF LOGARITHMIC CYCLE LENGTHS—

$$g = \frac{n_x}{n_x + n_y} d \quad \text{or} \quad h = \frac{n_y}{n_x + n_y} d$$

g = distance from z scale to x scale
h = distance from z scale to y scale
d = distance between x and y scales $= g + h$
n = length of logarithmic cycles designated by subscript

Laying out these distances and scale values as indicated in Fig. 6 gives the desired construction.

CONSTRUCTION METHOD TO GET THE SIZE OF z SCALE LOGARITHMIC CYCLE, n_z *in Fig. 6*—Draw lines connecting the upper values of x and y scales and between the lower end of these scales. The corresponding values of z for the top and bottom of the z scale can then be determined from the equation $xy = z$. The z scale is subdivided logarithmically between these two z values.

FORMULA METHOD FOR DETERMINING VALUE OF n_z—

$$n_z = \frac{h}{d} n_x = \frac{g}{d} n_y = \frac{n_x n_y}{n_x + n_y}$$

A starting point on the z scale can be determined by selecting a value of z within its range and determining values of x and y which will satisfy the equation. A line drawn across the three scales at the determined values of x and y will locate the selected value z on the z scale.

Assuming the following ranges in the equation $xy = z$ in Fig. 7; $x = 7$ to 18 and $y = 25$ to 40. Then z may vary from 175 to 720. The logarithmic scales for x and y are laid out on parallel lines taking most advantage of the available space.

To locate the z scale line by the construction method, any value of z could be used. If $z = 300$ is chosen, $x = 12$ when $y = 25$; and $x = 7.5$ when $y = 40$; therefore, $12 \times 25 = 300 = 7.5 \times 40$. Two intersecting lines through these values of x and y locate 300 on the z scale and a line is then drawn through the intersection parallel to the x and y scales.

If the nomogram were drawn on 8½x11 in. paper the length of the x scale cycle could be 25 in. since only approximately 0.4 of the cycle length is used. The length of the y cycle could be 50 in. since only 0.2 of the cycle length is used. Unless logarithmic scales of these lengths or longer are available for transferring or projecting the scale, a table of logarithms can be used in producing the scale on a uniformly divided scale of appropriate size (or on cross-section paper if the completed chart is to be traced for reproduction).

Assuming 25 and 50 in. cycles are used, $n_x = 25$ and $n_y = 50$.

$$\text{Then } g = \frac{25 \times 7}{25 + 50} = 2.333 \text{ in.} = \frac{d}{3} \text{ when } d = 7 \text{ in.}$$

The length of the z scale cycle as determined by formula is:

$$n_z = \frac{25 \times 50}{25 + 50} = 16.667 \text{ in.} = n_y/3 = 2n_x/3$$

Nomogram $x/y = z$

It is evident that the nomogram for $x\ y = z$ is the same as for $z/y = x$ *or* $z/x = y$. Therefore the nomogram shown in Fig. 7 can be applied to any of these three equations.

Calling the z scale the x scale and the x scale the z scale makes this nomogram apply to the equation $x/y = z$. However, it is conventional to have the z scale between the x and y scales. For the equation $x/y = z$ this can be accomplished by construction of the nomogram as in Fig. 7, but making the x and y scales ascend in opposite directions. The z scale then ascends in the same direction as the x scale.

When nomograms are made for $x^a\ y^b = z^c$ and $x^a/y^b = z^c$, as explained for Figs. 6 and 7, the constants

a, b, and c can be ignored if the construction method is used. The process is identical with that for $x\ y = z$ *and* $x/y = z$. Variations in the formulas allow for the constants. (Compare with the variations for $a\ x + b\ y = c\ z$ under the discussion for Fig. 2 nomogram.)

The location of z scale is:

$$\frac{g}{d} = \frac{n_x/a}{n_x/a + n_y/b}$$

The size of z scale logarithmic cycle

$$\frac{n_z}{c} = \frac{hn_x}{da} = \frac{gn_y}{db} = \frac{(n_x/a)\ (n_y/b)}{n_x/a + n_y/b} = \frac{n_x n_y}{bn_x + an_y}$$

Nomogram for $f_1(x)f_2(y) = f_3(z)$ *and* $f_1(x)/f_2(y) = f_3(z)$

All equations considered so far may still be treated as special examples of the $f_1(x) + f_2(y) = f_3(z)$ equation containing functions of x, y and z symbols.

Because of the greater frequency of applications of multiplication and division equations, it is advisable to use the shortcut of direct application of the logarithmic scale. Reproduction is facilitated by the use of standard logarithmic scales which are available in many sizes on logarithmic graph paper.

The procedure for making a nomogram of function scales is identical in every respect with the procedure for the simple formula, $x\ y = z$, except in the replacement of the logarithmic scales of the functions with scales of the variable.

The bevel gear formula used in Fig. 5 may be placed in chart form more simply by using function scales. The equation can be written:

$$S/(1 - M + M^2/3) = X$$

The limits of S are 0.1 to 0.65 and for M are 0.05 to 1.3. X varies from 0.1 to 2.50, and $f(M)$ from 0.263 to 0.950 $[f(M) = (1 - M + M^2/3)]$.

If logarithmic scales of the functions of S and M are laid out as indicated in the procedure for the equation $x/y = z$, the result will be as indicated by the function scales in Fig. 8. The function scales for S and for X are in this instance the same as for S and X, and therefore need not be replaced. A table of values of M and $f(M)$ will be as follows:

M	$f(M)$	M	$f(M)$
0.1	0.9033	0.7	0.4633
0.2	0.8133	0.8	0.4133
0.3	0.7300	0.9	0.3700
0.4	0.6533	1.0	0.3333
0.5	0.5833	1.1	0.3033
0.6	0.5200	1.2	0.2800
		1.3	0.2633

Nomogram for $f_1(x)+f_2(y)+f_3(z)+\ldots = f_4(w)$ *and for* $f_1(x) \cdot f_2(y) \cdot f_3(z) \cdot \ldots = f_4(w)$

Any equation of four or more variables may be split up into two or more equations of three variables and solved step by step until the final result is obtained. Thus, the equation $R = 0.0194\ W\ S\ V^2$ may be divided into two equations by introducing another variable, X, which becomes a factor common to both equations. Some transpositions of the equation $R = 0.0194\ W\ S\ V^2$ are given in the Table I with diagrams, convenient as a mental picture in visualizing the most desirable setup. Arrows indicate the direction of ascension of the function scales. (The direction of the V scale in Equation 6 in Table I is opposite to the $f(V)$ scale because $f(V) = 1/V^2$.) The constant may be attached to any variable.

The diagrams in Table I are of use only as relating the scale positions but not the actual distances between the scales.

The two equations so formed by introduction of the variable X, contain only three variables and may be treated by the rules previously laid out for the corresponding form, except when the X scale line is between the other two variables in the second equation. In this instance the size of the X scale has been determined from the first equation and must be of the same value in the second equation. Therefore it is necessary to begin with the predetermined sizes of one outside scale and the intermediate scale instead of the two outside scales.

It requires careful planning for a nomogram of this kind to get the greatest advantage of a given size of sheet. For example, in Fig. 9, if the ranges of the four variables are such that two variables require three cycles of logarithmic scale and the other two variables require only one cycle of logarithmic scale, a grouping of the first two variables with X will produce a small scale cycle of X. The single cycle scales are therefore only one-third as long as the space allows. But if a three-cycle and a one-cycle scale are combined in an equation with X, the remaining equation will be similar and all four scales will be the full length of the paper as in Fig. 10. If the single cycle scales of Fig. 9 had been drawn first to as large a scale as the space allows, the three-cycle scales would have overrun the standard logarithmic paper.

The easiest construction results when the X scale line is between the other two in the first equation and one of the outside scales in the second equation. The resulting nomogram may not be as economical of space but is satisfactory in most instances.

To use the largest possible scales in the second equation, those variables must be coupled in the first equation which will give the largest scale for X. Other more important considerations, however, may influence the grouping.

Nomogram for $T/P = Rb = X$

An example, as in Fig. 11, will better illustrate the type in which the X or reference line is between the other two variables in both equations. Equation for maximum band pressure of brakes is $P = T/Rb$ where

P = max. band pressure = 40 to 400 lb. per sq.in.

T = tension or compression = 250 to 15,000 lb.

R = radius of the drum = 4 to 20 in.

b = width of the band = 1.5 to 6 in.

In this instance it is considered more important that

TABLE I

Possible Arrangement	Equation R
1. $WS = X = R/0.0194\ V^2$	
2. $WV^2 = X = R/0.0194\ S$	
3. $WS = X$ and $0.0194\ V^2X = R$	
4. $R/W = X$ and $X/V^2 = 0.0194\ S$	
5. $R/X = 0.0194\ W$ and $SV^2 = X$	
6. $R/X = S$ and $0.0194\ W/X = 1/V^2$	
7. $X/S = W$ and $0.0194\ XV^2 = R$	

FIG. 6
Monogram for xy = z using logarithmic scales. The scale sizes are independent of each other

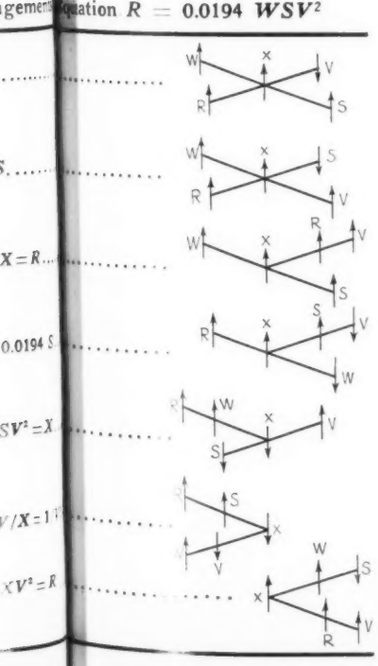
FIG. 7
The chart for the equation xy = z can also be applied to a transposition of this formula, namely z/y = x or z/x = y

FIG. 8
Simplified form of laying out bevel gear nomogram. Logarithmic scales are used for functions of S and M

FIG. 9
Nomogram for four variables as for equation, xy = uv

FIG. 10
When the equation shown in Fig. 9 is in the form of x/u = v/y, all scale sizes can be drawn as large as the space allows

FIG. 11
Chart layout of equation for maximum band pressure of a brake. Related variables as dimensions or pressures are coupled



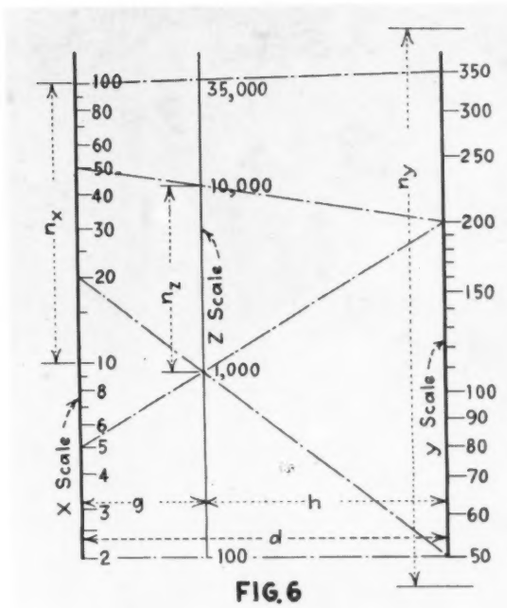


FIG. 6

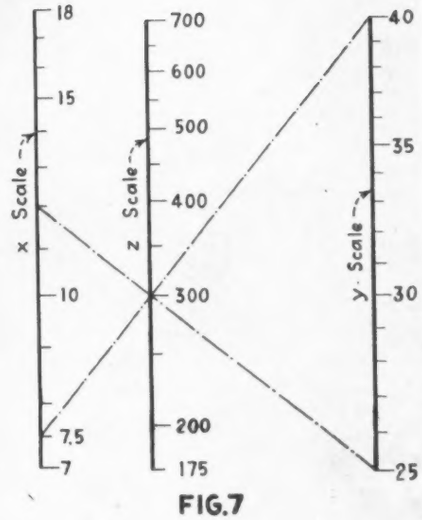


FIG. 7

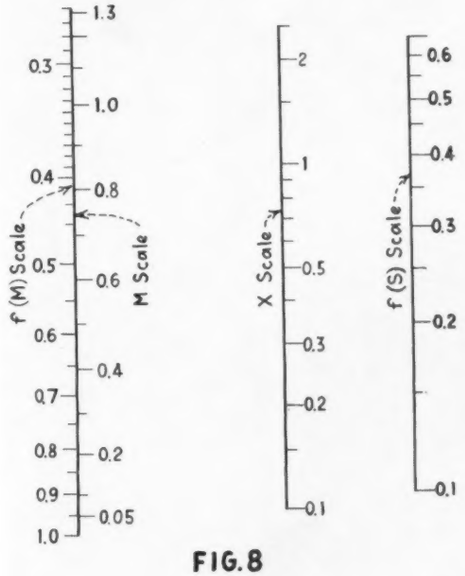


FIG. 8

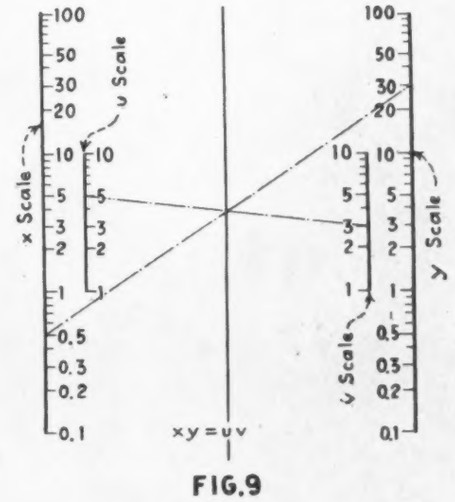


FIG. 9

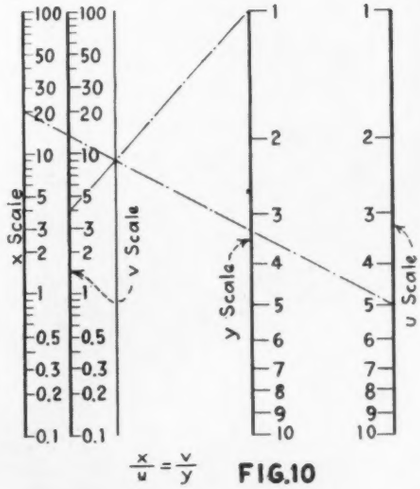


FIG. 10

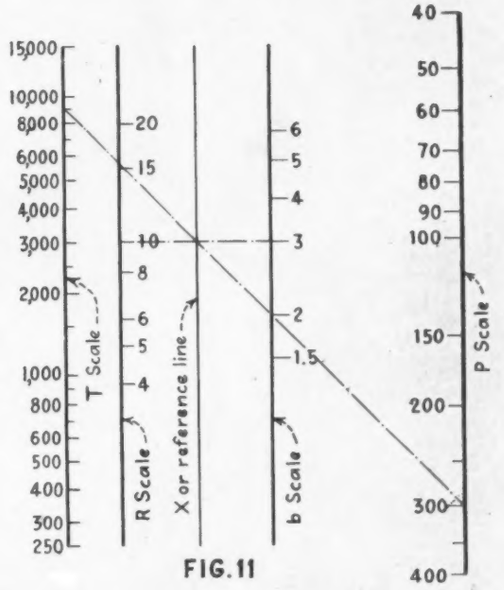


FIG. 11

R and b be coupled in one of the equations than that the greatest advantage be taken of the space, since both are dimensions. The remaining two variables also are more nearly related. The data is conveniently tabulated when the equation is written $T/P = Rb = X$.

	Limits	Cycles	Max. Cycle Length (a)	Actual Cycle Length
P	40 to 400	1 0	10 in.	10 in.
T	250 to 15,000	1 8	5 5 in.	5 in. (b)
R	4 to 20	0 7	14 in.	6 67 in.
b	1 5 to 6	0 6	16 6 in.	6 67 in.
$X = (T/P)$	0 6 to 375	2 8		3 33 in.
$X = (Rb)$	6 to 120			3 33 in.

(a) Product of max. cycle length × cycles = 10 in.
(b) Could be 5.5 in. Standard log paper has a 5 in. cycle.
For $X = (T/P)$, distance $g_1 = d_1/3$
For $X = (Rb)$, distance $g_2 = d_2/3$

The R and b scales had to be shortened to keep the P and T scales within the limits of the paper. The number of possible scale lengths for R and b is unlimited: 6.67 and 6.67, 10 and 5, or 15 and 4.285. The first group, 6.67 and 6.67 was chosen because there is approximately the same percentage of a cycle in each scale. Simplicity of construction was also considered. Determination of these figures was from the equation:

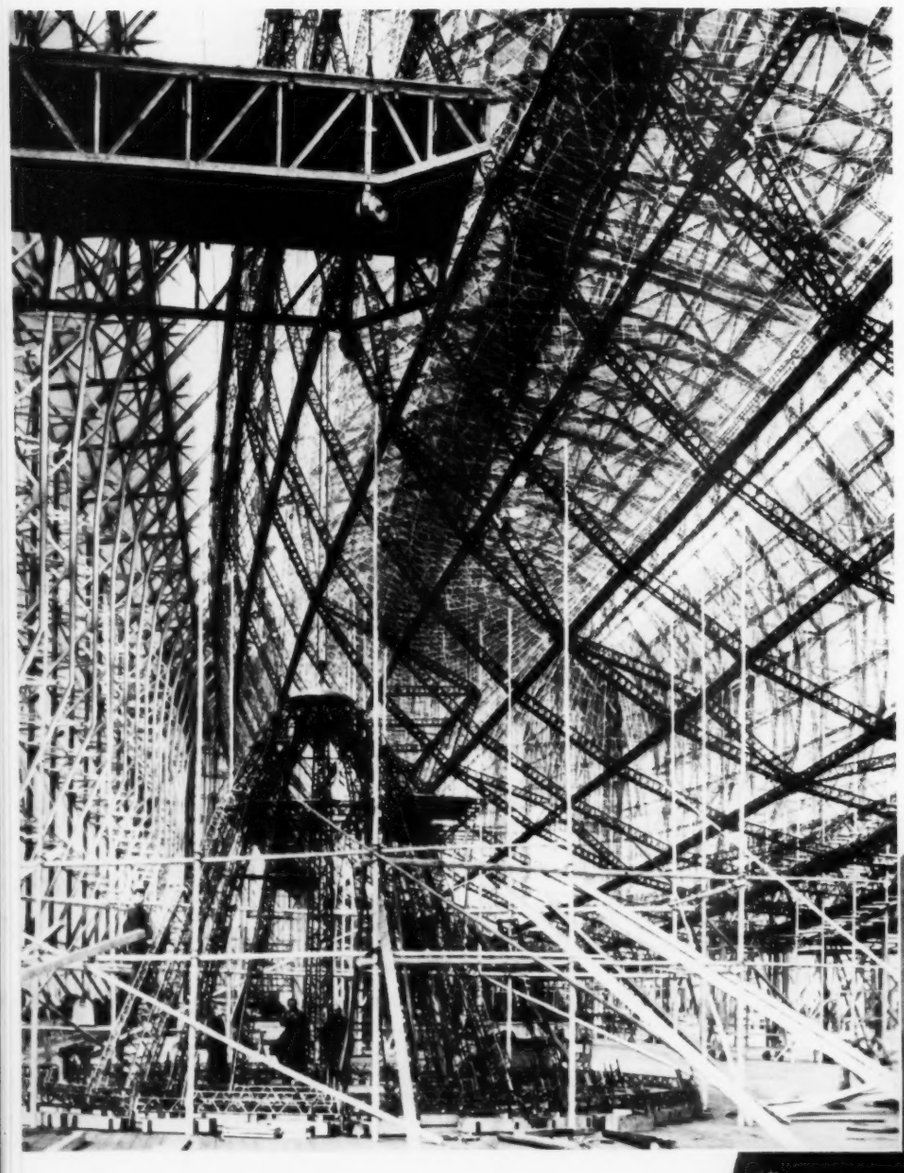
$$n_z = \frac{n_x n_y}{n_x + n_y} = \frac{10 \times 5}{10 + 5} = \frac{10}{3}$$

rearranged as a solution for n_x

$$n_x = \frac{n_z n_y}{n_y - n_z} = \frac{\frac{10}{3} \times \frac{20}{3}}{\frac{20}{3} - \frac{10}{3}} = \frac{\frac{200}{9}}{\frac{10}{3}} = \frac{20}{3}$$

and from the simple rule that when the outside function scales are equal, they are double the length of the middle function. Fig. 11 shows the framework of the completed nomogram.

Five variables in the equation simply add another reference line. Each additional variable adds another reference line and another equation of three variables.

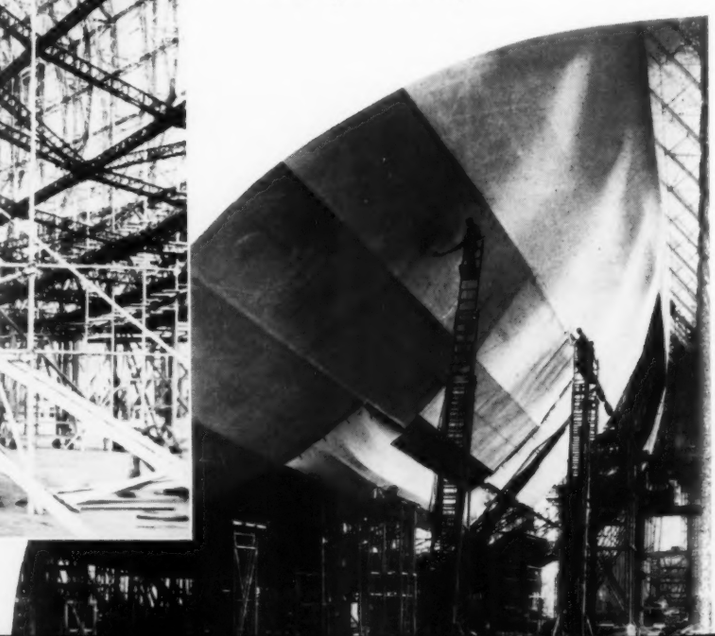


Largest Airship to Fly in February

NOW almost completed, the new German superliner of the air is scheduled to be finished this month. Designated as the LZ-129, this airship being built at Fredrichshafen, has a bulk of 7,000,000 cu.ft.—500,000 cu.ft. more than the ill-fated Akron and Macon, and almost twice as large as the Graf Zeppelin. The LZ-129 is 815 ft. long.

Better design and more attention to reducing wind resistance will give the new ship a top speed of about 85 m.p.h.. Five diesel engines with a total output of 4,200 hp. will operate on fuel oil.

The framework is entirely of aluminum alloy similar to that developed by the Goodyear Corporation at Akron, Ohio. Gas cells are being made from a single layer of sheet cellulose material. The outside cover is painted with a special aluminum paint which has a high reflectivity.



Metered gravity feed at four points along the main bearing in Skinner Unaflow steam engines

Lubrication of Sleeve Bearings

PART III

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Research Engineer, The Bunting Brass & Bronze Company

AS MACHINERY speeds increased and bearing loads became heavier, more attention had to be given to lubrication. It was not the cost of replacing burned-out bearings, but the loss entailed by machine idleness that created the demand for more dependable lubricating means. In large machines with many points requiring lubrication, the cost of manual lubrication for each individual bearing had become a serious factor. Certainty of lubrication, a minimum of dependence upon manual attention, and economy in the consumption of lubricant were desired. This led to the development of centralized lubricating systems.

Steam engines, turbines and diesel engines were the first to make extensive use of one form or another of centralized lubricating systems. In its simplest form the central lubricator was a small reservoir holding a pint or more of oil. Wicks dipping into the oil fed it to individual tubes that led to the various bearings. From this simple beginning there gradually developed the three modern types of centralized lubricating systems. These are: (1) Mechanical lubricators consisting of a reservoir and a pump which forces the lubricant through individual pipes to each of the bearings. (2) Gravity circulating systems wherein oil is pumped from a drain tank at the bottom to a storage tank at the top of the machine. Oil feeds by gravity through pipes to each of the bearings and then drains back to the lower tank. (3) The pressure circulating system wherein oil from a reservoir is pumped to the various bearings from which it drains back.

Mechanical lubricators may be manually operated, motor driven, or driven by the machine to which they are attached. The oil or grease is not circulated, the lubricator being adjusted so that the minimum amount of lubricant required will be fed to the bearing and consumed.

In the gravity circulating system an adjustable sight feed may be provided at each bearing. After passing through the bearing, the oil is filtered and then drains back to the lower reservoir. Or the filter may be placed in the line between the pump and the upper reservoir. This is the simplest type of circulating system and costs least. But it can only be used where all of the points requiring lubrication are readily accessible by gravity feed. It has the disadvantages that the oil is not supplied at a positive pressure and that pressure at the points lubricated will vary with their vertical location. It is relatively difficult to regulate the supply of lubricant fed to the various points, although sight feeds be used to indicate the rate of oil supply.

Pressure circulating systems may be had in a great variety. Oil filters are used almost always in order to assure a supply of clean oil to the bearings. In engines, turbines and machines where the oil gets heated, an oil cooler is usually used. Pressure switches controlling an electric circuit are sometimes used to give a signal or stop the machine if lubrication fails. Low pressure because of an inadequate supply of oil in the reservoir, broken pump or other reason will cause the pressure switch to operate a signal or stop the machine, or both. Excessive pressure because of a clogged pipe or other reason will similarly stop the machine.

In most machines wherein the oil pump is motor driven the control for the oil pump motor is interlocked with the machine driving motor. With this type of control the main motor cannot be started until the oil pump motor is running and furnishing an adequate supply of oil to the bearings. Similarly, stoppage of the oil pump motor shuts down the main motor.

Whether the oil pump in pressure circulating systems is to be driven by a separate motor or by the machine to

Mechanical lubricator delivering oil to important bearings through individual pipes

which it is attached depends on a number of factors none of which can be evaluated numerically. Where unit bearing pressures are high and there are long periods of idleness it is generally better to have a separate motor for driving the pump. This makes it possible to insure an abundant supply of oil at every bearing point before the machine is put in motion.

Where the pump is driven by the machine to which it is attached, some time will elapse after the machine has been started before the oil pressure has been built up. If high unit bearing pressure has squeezed out most of the oil during the period of machine idleness, the bearing might score, burn out or seize before the circulated lubricating oil reaches the bearing. Much will also depend on the material of the bearing. If the pump is to be driven by the machine, a bearing alloy with high non-scoring properties should be used.

Machine tools, steam turbines and diesel engines usually have an independent motor driven pump. Light machines usually have the oil pump driven by a moving element of the machine.

Oil pumps are ordinarily designed so that the volume discharged can be adjusted readily. This is done in several ways. The speed reduction between direct-connected driving motor and pump may be adjustable, or the size of sheaves or sprockets in V-belt or chain driven pumps may be changed. In other types discharge volume may be adjusted by changing the stroke of the plunger, or similar methods.

Main lines from the oil pump to the various headers or bearings may be $\frac{1}{4}$ in. or $\frac{5}{16}$ in. O.D. copper tubing. Brass tubing is sometimes preferred because it is stronger. Some designers prefer flexible tubing because it is easier to install and is more resistant to fatigue failure caused by vibrations. The occurrence of such failures in airplane engines led to the adoption of flexible tubing at all points where the pipe would be subjected to vibrations. In machinery where the vibrations are not too severe, and brass or copper tubing is used, it is formed into a spiral or helix at points of vibration.

Breakage of tubing caused by fatigue is most likely to occur at or near the fittings. The abrupt change in the rigidity or stiffness of the tubing at this point causes a concentration of the stresses created by the vibrations. For this reason some fittings are made with tapered ends, giving a gradual change in stiffness.

In addition to flexible tubing, oil resisting hose made or lined with Thiokol, Duprene or Koreseal or similar synthetic rubber-like materials are also available. These are specially suited for flexible connections between stationary and moving elements of the machine. Short lengths of such hose are sometimes used at the connection to the pump and other points in order to isolate the rigid tubing from vibrations.

Tubing delivering oil to the bearings should also be protected from extremes in temperature. In steam or diesel engines oil supply pipes must be kept away from steam lines or exhaust pipes to prevent thinning of the oil. Not only does the oil temperature affect its viscosity and therefore the load-carrying capacity of the oil film, but the lubricant also has a cooling function to perform in the bearing. On the other hand, extremely low oil temperatures will result in sluggish flow and deficient lubrication. The exact oil temperature needed is not critical, however.

As explained in the description of oil circulating systems for automobile engines (January *P. E.*) it is necessary to have a choke at each bearing to prevent the possibility of a free flow of oil through worn bearings or where clearances are large. In the various pressure circulating systems now available these "chokes" may be drip plugs, metering valves or feeders in a variety of designs. Some are equipped with sights for a quick visual check to see if the system is operating properly. Most of them are readily adjustable to change the rate of feed.

Single feeders or drip plugs may be used at each bearing. But usually it is better to group the feeders to supply groups of bearings. The "gang" of feeders is located where most convenient for observing the sight feeds and adjusting the feed rates.

Simplifying Helical Spring Design

Time saving method for designing springs when load-deflection rate can be used as a starting point

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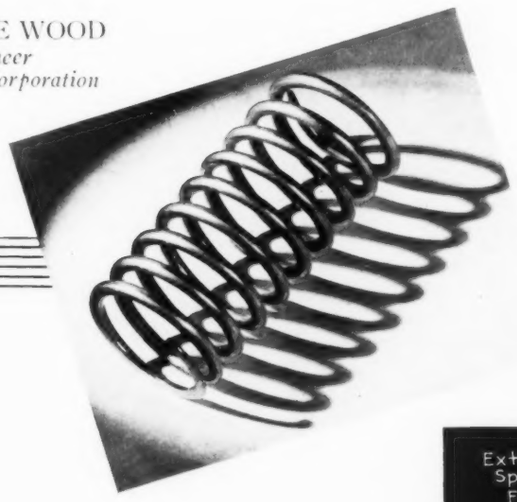
Chief Engineer
General Springs Corporation



IN THE design of springs, too much emphasis is often placed on the economical use of material, and not enough attention is given to safety against overstressing. Hitting the deadline between safety and economy of material with great accuracy is of less importance than equal accuracy in predicting loads and deflections. By adopting a more liberal design for most springs, design time can be saved and procedure simplified.

In the usual spring application it is not the maximum load corresponding to the safe maximum fiber stress that is essential in starting the design. The starting point ordinarily is the load at one or more given positions in the travel of a moving member in the mechanism to which the spring is attached. The design of the mechanism usually determines where the ends of the spring are located, and hence establishes the length of the spring.

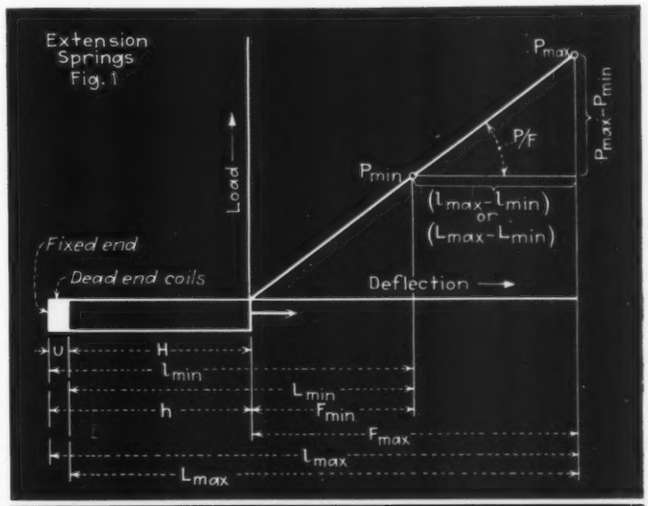
In extension helical springs the maximum operating length determines the sum of the deflection and solid length, while in compression springs the free length determines the sum of the maximum deflection and solid length. For such springs, design procedure can be simplified by starting with the load-deflection rate.

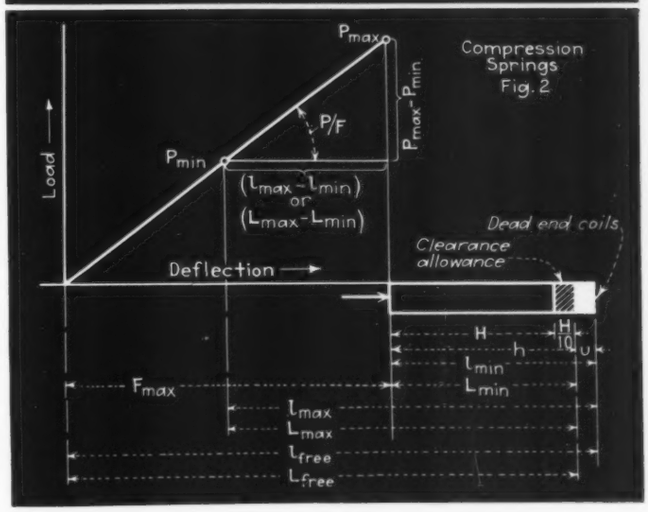
If more than one operating load is fixed during the travel, the load-deflection rate is determined, and this factor then must be the starting point in the design. If only one load is fixed, resonant vibration effects may determine the load deflection rate; if not, it is only necessary to choose a deflection that is within a safe proportion of the live solid length.

In the following formulas based on loads and lengths, all quantities are whole numbers, none of which depend on the number of coils. The load $P_{max.}$ corresponds to a fiber stress of 50,000 lb. per sq.in. corrected for the Wahl factor, and the load $P_{min.}$ is any load less than the maximum. The maximum spring length $L_{max.}$ for compression springs is the live length of the spring (center-to-center distance between outer coils) corresponding to the minimum load $P_{min.}$ For extension springs $L_{max.}$ is the live length under the maximum load.

Minimum spring length, $L_{min.}$ for compression springs is the live length of the spring plus 10 per cent for clearance, and corresponds to the maximum load $P_{max.}$ For extension springs $L_{min.}$ is the live length corresponding to $P_{min.}$ All terms

Figs. 1 and 2—Load deflection diagrams for extension and compression springs





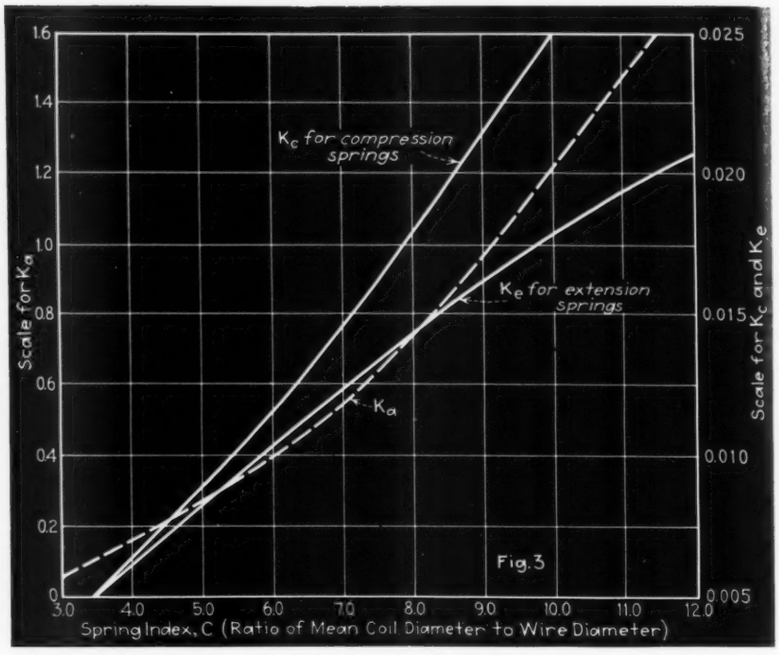


Fig. 3 — Constants used in Equations (6), (11) and (12) for springs designed for a given load-deflection rate

are in pounds and inches. The value used for the torsional modulus G is 11,470,000 lb. per sq.in. for steel. The Wahl factor K depends on the spring index C, or ratio of D, the mean coil diameter to d, the wire diameter.

In terms of required maximum and minimum total lengths L, or effective lengths l, the load-deflection rate, which is the load P divided by the deflection F, is given by,

$$\frac{P}{F} = \frac{P_{max} - P_{min}}{L_{max} - L_{min}} = \frac{P_{max} - P_{min}}{l_{max} - l_{min}} \qquad (1)$$

In determining the design, the load deflection rate calculated from Equation (1) is expressed as a function of the torsional modulus G, the spring index C, the wire diameter d, and the live solid height H, as follows:

$$\frac{P}{F} = \frac{Gd^2}{8C^3 H} = K_b \frac{d^2}{H} \qquad (2)$$

To simplify design calculations, the torsional modulus and the spring index are combined in the constant K_b. For compression springs, using the notation in Fig. 2 and allowing 10 per cent for clearance between coils, the minimum total length is given by,

$$L_{min} = l_{min} - U = 1.1 H \qquad (3)$$

$$H = \frac{l_{min} - U}{1.1} \qquad (4)$$

For extension springs using the notation given in Fig. 1, the maximum length is,

$$L_{max} = l_{max} - U = H + F_{max} \qquad (5)$$

$$F_{max} = \frac{\pi SC^2 H}{GK} = \frac{\pi \times 50{,}000\, C^2 H}{11{,}470{,}000\, K} = K_a H \qquad (6)$$

In Equation (6) the factor K_a includes the Wahl factor K, the spring index, working stress and modulus. Substituting the value of $F_{max.}$ in Equation (5), the corresponding length becomes,

$$L_{max} = H(1 + K_a) \qquad (7)$$

$$H = \frac{l_{max} - U}{1 + K_a} \qquad (8)$$

The length U is that part of the spring length represented by dead end coils. Substituting the values of H given in Equation (4) and (8) into Equation (2), the wire diameter d for a compression spring is given by,

$$d^2 = \frac{(l_{min} - U)}{1.1 K_b} \times \frac{P_{max} - P_{min}}{l_{max} - l_{min}} \qquad (9)$$

For extension springs the wire diameter is given by the relation,

$$d^2 = \frac{(l_{max} - U)}{K_b (1 + K_a)} \times \frac{P_{max} - P_{min}}{l_{max} - l_{min}} \qquad (10)$$

For a given material the value of K_b depends only on the spring index. Thus, Equation (9) for compression springs can be simplified by introducing a new constant K_c expressed in terms of the first power of the wire diameter d. Since the right-hand term in Equation (9) is equal to the load-deflection rate, the wire diameter for compression springs is given directly by

$$d = K_c \sqrt{(l_{min} - U) P F} \qquad (11)$$

In the same manner, Equation (10) for extension springs can be simplified by introducing a new constant K_e. For extension springs the wire diameter is,

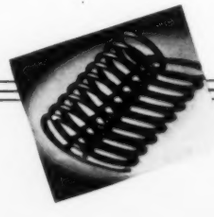
$$d = K_e \sqrt{(l_{max} - U) P F} \qquad (12)$$

Both K_c and K_e can be expressed in terms of the spring index C. For steel springs with a working stress of 50,000 lb. per sq.in., and a torsional modulus of 11,470,000 lb. per sq.in., the constants have values given in Fig. 3 by the curves.

When maximum and minimum loads and lengths are given, steel extension and compression springs can be quickly designed, using Equations (9) or (10) and the constants given by the curves in Fig. 3. First, the load-deflection ratio is found using Equation (1). Then, K_a is determined as follows. Since $P_{max.}$ equals $F_{max.} \times P/F$ and $F_{max.}$ is equal to $K_a H$, the maximum load is,

$$P_{max} = K_a H P F \qquad (13)$$

For compression springs $H = (l_{min} - U)/1.1$ and for extension springs $H = (l_{max.} - U)/(1 + K_a)$.



Substituting these relations for H in Equation (13) the values for K_a required by the design are,

$$\text{(Compression)}\quad K_s = \frac{1.1}{\dfrac{P}{F} \times \dfrac{l_{min} - U}{P_{max}}} \qquad (14)$$

$$\text{(Extension)}\quad K_s = \frac{1.0}{\left(\dfrac{P}{F} \times \dfrac{l_{max} - U}{P_{max}}\right) - 1.0} \qquad (15)$$

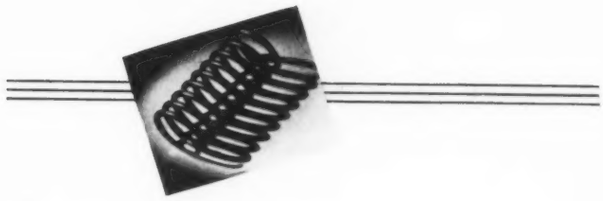
The dead length U is estimated and the value of K_a is calculated from Equation (14) or (15). For compression springs, the dead length can be taken as $2\ d$, while for extension springs the type of end determines the length of dead coils. From the calculated value of K_a, the curve in Fig. 3 gives the spring index required.

Using the spring index now determined, values of K_s or K, also are read from Fig. 3 and substituted in Equations (11) or (12) giving the required wire diameter directly.

Mean coil diameter is obtained from the spring index, and the number of live coils N_L is given by,

$$\text{(Compression)}\quad N_L = \frac{H}{d} = \frac{l_{min} - U}{1.1\,d}$$

$$\text{(Extension)}\quad N_L = \frac{H}{d} = \frac{l_{max} - U}{I + K_s}$$



The total number of coils N_t for compression springs is,

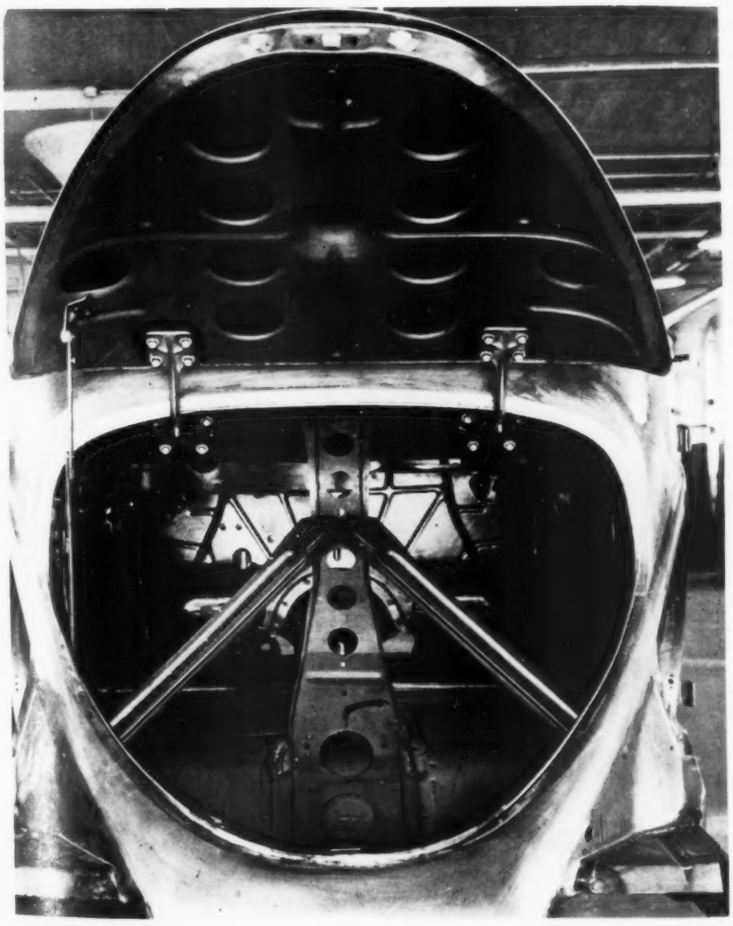
$$N_t = N_L + U/d$$

Free or overall length l_f is obtained from the relations,

$$\text{(Compression)}\ l_f = l_{min} + K_s N_L d$$
$$\text{(Extension)}\ l_f = N_L d + U$$

All of the computations of number of coils, and the free and overall length are operations involving factors already given or previously determined.

Curves in Fig. 3 are for steel springs with a maximum working stress of 50,000 lb. per sq.in. There are some applications in which it might be desirable to use a higher stress to reduce material cost or decrease weight. On the other hand, most compression and extension springs do not require the minimum possible weight, and the use of a little more metal in a few springs with consequent saving in designing time is well worth while. For such springs the procedure outlined above has the advantage of leading to a direct solution for the wire diameter without cut and try methods. Also the estimate of dead length is in the hands of the designer and is not buried in the formulas.



Lincoln-Zephyr Body Assembled by Welding

WITHOUT modern developments in welding the new type of body construction used in the Lincoln-Zephyr would hardly be possible. The entire body made of nearly 300 separate pieces of metal is welded into a one-piece structure. Nearly all types of welding are used in building up the three principal units—the floor section, front end, and rear—and in welding these three units together. The steel roof is welded to the rear section.

All structural loads are carried in the body itself since no chassis is used. Therefore all parts must be permanently and rigidly attached. Approximately 3,800 welds are required with more than 3,400 of them made by portable resistance welders. Arc welding is also used, principally for tacking parts together and for joining braces and similar parts to channels. Thin metal sections are welded with oxyacetylene torches which are also used in filling seams.

Special butt welding machines are used for long seams between the top and the front and rear quarters. Panels in the rear section, shown in the illustration, are held in jigs and fused into one piece in a large resistance welding machine.

An assortment of miscellaneous screw-machine parts made from aluminum alloys. One of the recent developments has been free-cutting aluminum alloys which permit high-speed production of such screw-machine parts

ALLOYS of ALUMINUM

.... And Their Uses

PAUL V. FARAGHER
Aluminum Company of America

Range of properties of commonly used aluminum alloys, the forms in which they are available, their fabrication and heat-treatments and purposes for which they are being used . . . .

FIFTY years ago, the twenty-third of this month, when the electrolytic process for its production was discovered by Charles Martin Hall, aluminum was simply the metallic element of that name. To the average man today, aluminum includes indiscriminately the whole group of aluminum alloys used in the construction of the "aluminum" train, or truck, or bus or airplane or bridge, which have made news in the daily press. To the engineer, however, aluminum is each one of the alloys in the form and temper best suited for the particular design.

Aluminum of commercial purity has many desirable qualities; it is light, it responds readily to commercial fabricating processes, it resists the attack of the atmosphere, including both industrial locations with their sulphurous gases and the sea coast with its salt air, it is a good conductor of both heat and electricity, and a good reflector of light, heat and other forms of radiant energy. However, the pure metal is comparatively soft and the use of aluminum in structural applications has had to await the modern developments of the alloying art.

The elements most commonly used in the commercial aluminum alloys are copper, silicon, magnesium, manganese, iron, nickel, zinc and chromium. Small amounts of other elements are present in some of the alloys, but the greater number contain some combination of the hardeners just mentioned. While the purpose of alloying is, primarily, to increase the strength, practically all of the properties of the resulting product differ in some degree from those of the parent metal. In some of the qualities there may be improvement, in others there may be a sacrifice, while in some the change may not be significant in relation to the commercial applications of the metal.

Tables I and II give the approximate composition of the principal wrought and sand casting aluminum alloys.

The outstanding quality of lightness undergoes little change in the commercial alloys of aluminum. A few, in which silicon or magnesium is the principal alloying element, are slightly lighter than pure aluminum. In none of the wrought alloys is the increase in specific gravity greater than three per cent. In a few of the cast alloys the increase is somewhat more; but in none which are now in common use is it as much as ten per cent.

The ductility and forming qualities undergo some loss, in every case, as compared with pure aluminum, although many of the alloys differ but little in this quality from pure aluminum.

Similarly, the resistance to corrosive attack is in no case improved over that of the parent metal although some of the alloys show no decrease in this property. All of the commercial alloys of aluminum may be classed as materials resistant to corrosion, but for some uses, the difference in this quality between two different alloys may be sufficient to dictate the choice between them.

Alloy Composition

In like manner, comparisons might be drawn for the other properties of the metal. It is sufficient to state, however, that no two of the many commercial alloys among which the user must choose, are identical in all their qualities. Some of the alloys have practically identical tensile properties, but differ in other qualities sufficiently to justify the retention of both alloys in commercial production.

By alloying alone, the strength of aluminum may be doubled; and by cold working the resulting product, the strength may be further increased to somewhat less than twice that of the alloy in the annealed state.

The combination of heat-treatment processes with proper selection of alloy composition, makes possible much higher strengths, and without the same sacrifice in ductility which results from cold working the alloy to strain-harden it. The first of the heat-treatable alloys of aluminum called duralumin by its inventor, Alfred Wilm, a German metallurgist, was brought into production in time to be of some service in the construction of aircraft used in the World War.

The development of this new class of materials, light like the parent metal, but strong as steel, opened to

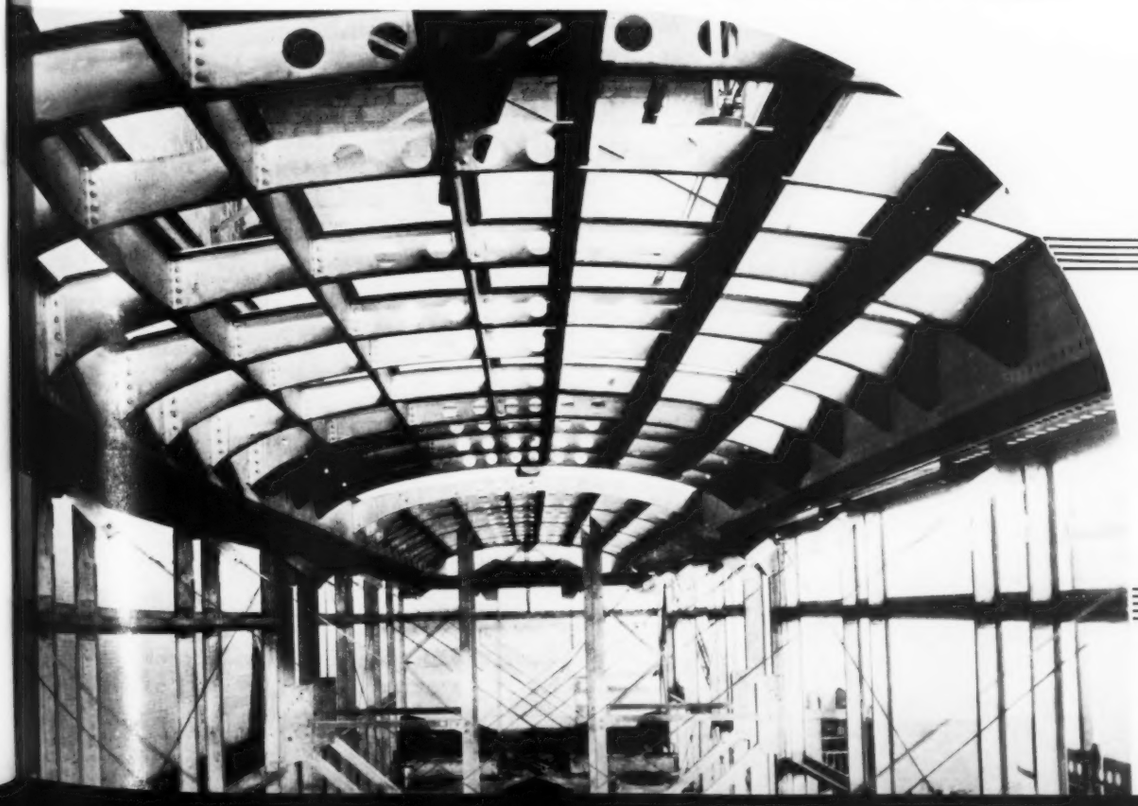
Table I—Nominal Composition of Wrought Aluminum Alloys‡

Alloy	Per Cent of Alloying Elements. Aluminum and Normal Impurities Constitute Remainder						
	Copper	Silicon	Manganese	Magnesium	Zinc	Nickel	Chromium
2S	.....	.....	.....	.....	.....	.....	.....
3S	.....	.....	1.25	.....	.....	.....	.....
4S	.....	.....	1.25	1.0	.....	.....	.....
17S	4.0	.....	0.5	0.5	.....	.....	.....
A17S	2.5	.....	.....	0.3	.....	.....	.....
24S	4.2	.....	0.5	1.5	.....	.....	.....
27S	4.5	0.8	0.8	.....	.....	.....	.....
51S	.....	1.0	.....	0.6	.....	.....	.....
52S	.....	.....	.....	2.5	.....	.....	0.25
53S	.....	0.7	.....	1.25	.....	.....	0.25

Table II—Approximate Composition of Aluminum Sand Casting Alloys‡

Alloy	Per Cent of Alloying Elements. Aluminum and Normal Impurities Constitute Remainder						
	Copper	Iron	Silicon	Zinc	Magnesium	Nickel	Manganese
12	8.0	.....	.....	.....	.....	.....	.....
43	.....	.....	5.0	.....	.....	.....	.....
112	7.5	1.2	.....	1.5	.....	.....	.....
122	10.0	1.2	.....	.....	0.2	.....	.....
142	4.0	.....	.....	.....	1.5	2.0	.....
195	4.0	.....	.....	.....	.....	.....	.....
212	8.0	1.0	1.2	.....	.....	.....	.....
214	.....	.....	.....	.....	3.75	.....	.....
220	.....	.....	.....	.....	10.0	.....	.....
355	1.25	.....	5.0	.....	0.5	.....	.....
A355	1.4	.....	5.0	.....	0.5	0.75	0.75
356	.....	.....	7.0	.....	0.3	.....	.....

Heat-treatment symbols have been omitted since composition does not vary for different heat-treatment practices. ‡The compositions and/or heat treatment of many of these alloys are patented.



Roof construction of the all-aluminum streamlined train "Comet" of the New York, New Haven and Hartford Railroad. Note the transverse roof members with lightening holes and the large longitudinal corner members

Table III—Typical Mechanical Properties of Wrought Aluminum Alloys[1]

ALLOY AND TEMPER	TENSION				HARD-NESS	SHEAR	FA-TIGUE	ALLOY AND TEMPER	TENSION				HARD-NESS	SHEAR	FA-TIGUE
	Yield Strength[2] (Set = 0.2%) Lb. per sq. in.	Ultimate Strength Lb. per sq. in.	Elongation[3] Per cent in 2 in. Sheet Specimen (1/16 in. Thick)	Elongation[3] Per cent in 2 in. Round Specimen (0.505 in. Diam.)	Brinell 500 kg. 10 mm. Ball	Shearing Strength[4] Lb. per sq. in.	Endurance Limit[5] Lb. per sq. in.		Yield Strength[2] (Set = 0.2%) Lb. per sq. in.	Ultimate Strength Lb. per sq. in.	Elongation[3] Per cent in 2 in. Sheet Specimen (1/16 in. Thick)	Elongation[3] Per cent in 2 in. Round Specimen (0.505 in. Diam.)	Brinell 500 kg. 10 mm. Ball	Shearing Strength[4] Lb. per sq. in.	Endurance Limit[5] Lb. per sq. in.
2S-O	4,000	13,000	35	45	23	9,500	5,000	A17S-O	8,000	22,000	24	27	38	15,000	.....
2S-1/4H	13,000	15,000	12	25	28	10,000	6,000	A17S-T	24,000	43,000	24	27	70	25,000	13,500
2S-1/2H	14,000	17,000	9	20	32	11,000	7,000	24S-O	10,000	26,000	20	22	42	18,000	14,000
2S-3/4H	17,000	20,000	6	17	38	12,000	8,000	24S-T	43,000	65,000	20	22	105	40,000	14,500
2S-H	21,000	24,000	5	15	44	13,000	8,500	24S-RT	53,000	68,000	13	..	116	41,000	.....
3S-O	5,000	16,000	30	40	28	11,000	7,000	Alclad 24S-T	40,000	60,000	18	..	..	39,000	.....
3S-1/4H	15,000	18,000	10	20	35	12,000	8,000	Alclad 24S-RT	49,000	62,000	11	..	..	39,000	.....
3S-1/2H	18,000	21,000	8	16	40	14,000	9,000								
3S-3/4H	21,000	25,000	5	14	47	15,000	9,500	27S-T	50,000	60,000	9	11	115	37,000	13,000
3S-H	25,000	29,000	4	10	55	16,000	10,000								
								51S-O	6,000	16,000	30	35	28	11,000	6,500
4S-O	10,000	26,000	20	25	45	16,000	14,000	51S-W	20,000	35,000	24	30	64	24,000	10,500
4S-1/4H	25,000	31,000	6	15	55	17,000	14,500	51S-T	38,000	48,000	14	16	95	30,000	10,500
4S-1/2H	31,000	35,000	5	10	65	19,000	15,000								
4S-3/4H	35,000	39,000	3	7	73	21,000	15,500	52S-O	14,000	29,000	25	30	45	18,000	17,000
4S-H	38,000	42,000	3	5	80	23,000	16,000	52S-1/4H	26,000	34,000	12	18	62	20,000	18,000
								52S-1/2H	29,000	37,000	10	14	67	21,000	19,000
17S-O	10,000	26,000	20	22	45	18,000	11,000	52S-3/4H	34,000	39,000	8	10	74	23,000	20,000
17S-T	35,000	58,000	20	22	100	35,000	15,000	52S-H	36,000	41,000	7	8	85	24,000	20,500
17S-RT	46,000	61,000	13	..	110	36,000	.....								
								53S-O	7,000	16,000	25	35	26	11,000	7,500
Alclad 17S-T	32,000	55,000	18	..	..	32,000	.....	53S-W	20,000	33,000	22	30	65	22,000	10,000
Alclad 17S-RT	40,000	57,000	11	..	..	32,000	.....	53S-T	32,000	38,000	14	20	80	26,000	11,000

(1) Young's modulus of elasticity is approximately 10,300,000 pounds per square inch.

(2) Stress which produces a permanent set of 0.2 per cent of the initial gauge length. (American Society for Testing Materials Specification for Methods of Tension Testing, E 8-33.)

(3) Elongation values vary with the form and size of tension test specimen. Thin sheet has somewhat lower elongation than values for 1/16 inch sheet shown in table. Thicker material, from which standard round tension test specimens (0.505 inch diameter) are tested, may have lower elongation because of the effect of commercial flattening operations on this property.

(4) Single-shear strength values obtained from double-shear tests.

(5) Based on withstanding 500,000,000 cycles of completely reversed stress, using the R. R. Moore type of machine and specimen.

Table IV—Mechanical Properties of Sand Cast Aluminum Alloys [1]

ALLOYS	TENSION[3] Typical Values: Yield Strength[2] (Set = 0.2%) Lb. per sq. in.	TENSION[3] Typical Values: Ultimate Strength Lb. per sq. in.	TENSION[3] Typical Values: Elongation Per cent in 2 In.	TENSION[3] Min. Specification Values: Ultimate Strength Lb. per sq. in.	TENSION[3] Min. Specification Values: Elongation Per cent in 2 In.	COMPRESSION[6] Typical Values: Yield Strength[2] (Set = 0.2%) Lb. per sq. in.	COMPRESSION[6] Typical Values: Ultimate Strength Lb. per sq. in.	HARDNESS Typical Values: Brinell 500 kg. 10 mm. ball	SHEAR Typical Values: Shearing Strength[4] Lb. per sq. in.	FATIGUE Typical Values: Endurance Limit[5] Lb. per sq. in.	DENSITY Typical Values: Lb. per cu. in.
12 and 212	14,000	22,000	2.0	19,000	(7)	16,000	38,000	65	20,000	7,500	0.102
43	9,000	19,000	4.0	17,000	3.0	9,000	25,000	40	15,000	6,500	0.096
112	14,000	22,000	2.0	19,000	(7)	24,000	44,000	70	20,000	8,500	0.103
122-T2	21,000	25,000	1.0	23,000	(7)	33,500	54,000	75	25,500	9,500	0.106
122-T61	30,000	36,000	1.0	30,000	(7)	40,500	80,000	100	29,500		0.106
142	24,000	28,000	1.0	23,000	(7)	34,000	52,000	85	24,000	8,000	0.101
142-T61		37,000	0.5	32,000	(7)	58,000	70,000	100	32,000	8,000	0.101
142-T571	28,000	30,000	0.5	29,000	.....	30,000	77,000	85	27,000	8,000	0.101
195-T4	16,000	31,000	8.0	29,000	6.0	27,000	43,000	65	28,000	6,000	0.100
195-T6	22,000	36,000	4.0	32,000	3.0	29,000	48,000	80	30,000	6,500	0.100
195-T62	27,000	40,000	2.0	36,000	(7)	45,500	56,000	95	31,000	7,000	0.100
214	12,000	25,000	9.0	22,000	6.0	12,000	50,000	50	19,000	5,500	0.095
220-T4	26,000	44,000	13.0	40,000	11.0	23,500	72,500	75	33,500	7,500	0.092
355-T4	20,000	30,000	4.0	27,000	4.0	25,000	65,000	60	30,000	.....	0.097
355-T6	27,000	35,000	2.0	32,000	2.0	29,000	68,000	80	30,000	.....	0.097
355-T51	23,000	28,000	1.5	25,000	(7)	24,000	52,000	60	21,000	6,500	0.097
A355-T51	24,000	28,000	1.5	25,000	(7)	24,000	54,000	65	21,000	8,000	0.099
A355-T59	21,000	25,000	2.0	23,000	(7)	21,000	52,000	60	20,000	8,000	0.099
356-T4	16,000	28,000	6.0	26,000	5.0	16,000	46,000	55	22,000	.....	0.095
356-T6	22,000	32,000	4.0	30,000	3.0	21,000	48,000	70	23,000	8,000	0.095
356-T51	20,000	25,000	2.0	22,000	(7)	17,000	80,000	55	18,000	6,000	0.095

(1) Young's modulus of elasticity is approximately 10,300,000 pounds per square inch.

(2) Stress which produces a permanent set of 0.2 per cent of the initial gauge length. (American Society for Testing Materials Specification for Methods of Tension Testing, E 8-33.)

(3) Tension values determined from standard half inch diameter tensile test specimens individually cast in green sand molds and tested without machining off the surface.

(4) Single-shear strength values obtained from double-shear tests.

(5) Based on withstanding 500,000,000 cycles of completely reversed stress using the R. R. Moore type of machine and specimen.

(6) Results of tests on specimens having an l/r ratio of 16 to 20. All specimens failed by lateral bending.

(7) Not specified. The error in determining low elongations is comparable with the value being measured.

aluminum the field of structural applications. The alloy 17S, substantially the same as the original duralumin, was soon followed by other heat-treatable wrought alloys to meet the demands of the various types of service. In addition, heat-treated castings have come into general use extending the use of light castings into fields for which the older alloys, because of their lower properties, were not suited.

It is safe to say that no material possesses all properties in just the degree that might be desired. In general the choice involves some element of compromise. To reduce this element of compromise to the minimum, the number of alloys of aluminum in commercial production is rather large. Table III lists some of the wrought alloys, and Table IV the cast alloys which are in common use, together with their characteristic physical properties.

Designating Composition

In the system of nomenclature used by Aluminum Company of America, a number is used to designate a certain alloy composition, followed, in the case of the wrought alloys, by the letter S. Following the alloy number is the symbol which indicates its temper. For the alloys whose harder tempers are developed by strain-hardening, fractional symbols are used to indicate the tempers intermediate in strength between that of the soft or annealed temper (O), and that of the hard temper (H) which results from the maximum commercially practicable amount of cold working of the alloy. In general, the requirements for these intermediate tempers are met by the quarter-hard, half-hard and three-quarter-hard tempers, designated $-\frac{1}{4}$H, $-\frac{1}{2}$H and $-\frac{3}{4}$H respectively.

For the heat-treatable wrought alloys the letter T indicates complete heat-treatment. The letter W, used only with the alloys which require both solution and precipitation heat-treatment to develop their maximum strengths, indicates that only the former of these processes has been applied. The symbol –RT indicates that the alloy has been strain hardened after heat-treatment.

Where more than one heat-treatment is applicable to an alloy, particularly alloy castings, the various tempers may be indicated by a number following the letter T, such as 195-T4, 195-T6, etc. A letter preceding the alloy number represents a composition varying slightly from that of an older alloy, for example, 17S and A17S.

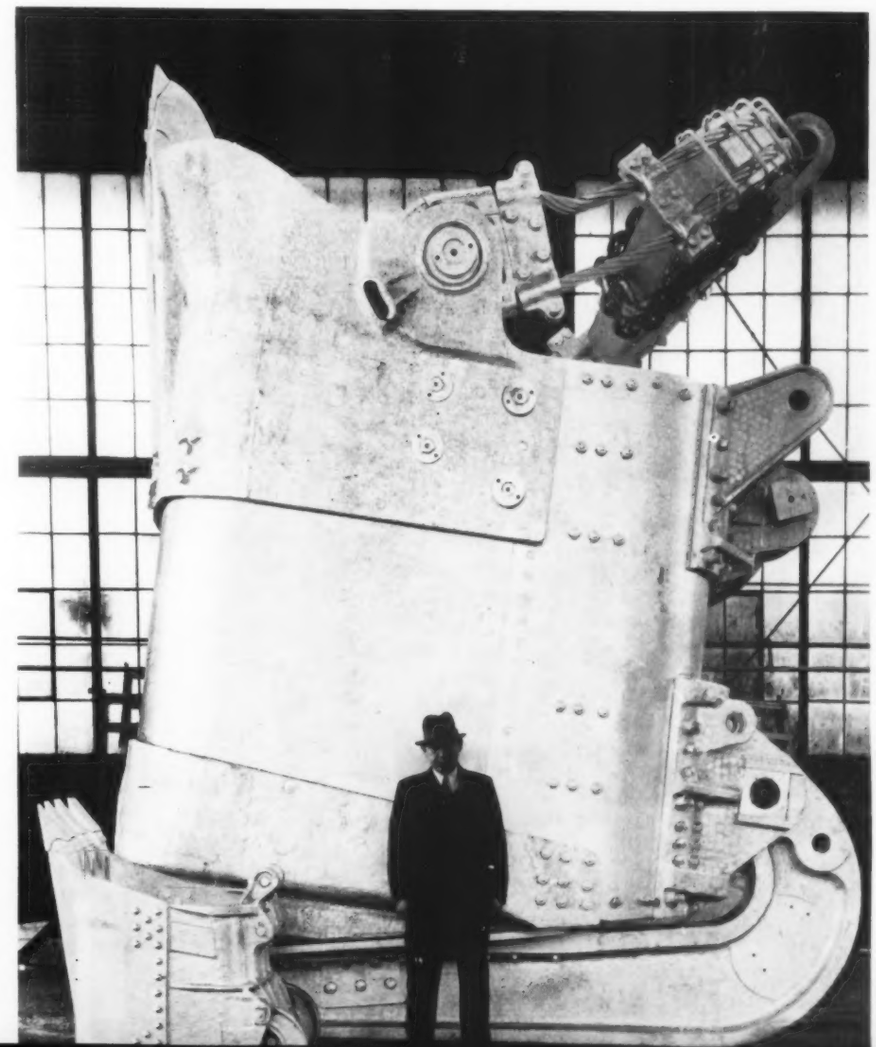
The class of wrought alloys which depends upon strain-hardening for the harder tempers includes 3S, 4S and 52S. Commercially pure aluminum, designated 2S, is also grouped with these alloys, although it is an alloy only by the presence of the impurities which normally occur in the metal.

These alloys are characterized by the fact that their resistance to corrosion is substantially the same as that of 2S. The newer alloy 52S is fully the equal of 2S and may be somewhat superior in this respect. In the form of sheet, wire and tubing, all the intermediate tempers as well as the soft and hard tempers are regularly produced. Plate is commonly rolled to size from hot ingots, hence the properties vary, depending upon the amount of cooling during the rolling process. Thick plate will have a tensile strength substantially the same as that of the soft temper, although the yield strength is generally higher than that of fully annealed material. Thinner plates may have strengths comparable with those of the quarter-hard temper of the alloys.

Similarly, bar and rod are rolled, and shapes are rolled or extruded from hot ingots and are, therefore, not regularly manufactured in the various definite tempers.

These products, which are produced directly from hot ingots, are normally supplied "as fabricated" since, for most purposes, accurate control of temper is not essential. Where higher strengths are required they are obtained by choosing a harder alloy (3S instead of 2S, 52S instead of 3S), or by the use of a heat-treatable alloy.

The mechanical properties of these alloys are adequate for a great variety of applications. Because of their greater ease of manufacture, their cost is less than that of the heat-treatable alloys. In their harder



A 32-cu.yd. shovel dipper made possible by the use of aluminum alloys

tempers, their yield strengths are comparable with those of some of the heat-treated alloys. They are, therefore, widely used.

For deep drawing and spinning operations, 2S is generally chosen. For less severe forming, 3S may be chosen with a gain in strength and stiffness of the part. Large tonnages of both of these alloys in the form of sheet are used for the manufacture of cooking utensils. In order that the finished article shall have the maximum strength and stiffness, the hardest temper of the alloy which will withstand the required forming operation is chosen. For some articles such as lids and trays which require only shallow draws, 3S–$\frac{3}{4}$H or even 3S–H may be used.

Mechanical Properties of 4S and 52S

The alloys 4S and 52S are intermediate in their mechanical properties between 2S and 3S, on the one hand, and the heat-treatable alloys, on the other. The tensile strengths are similar in corresponding tempers but 4S has a somewhat higher yield strength, which fact, coupled with its lower elongation, makes it rather more difficult to form than 52S. In the hard temper, the yield strength of 52S alloy is about the same as that of 17S–T. It does not have so high an ultimate strength as the latter heat-treated alloy, but for many applications in which rigidity and resistance to deformation are the chief requisites, this alloy is especially suited.

When more severe forming operations must be performed on the metal, a softer temper should be chosen. The cold working which it receives, while being formed, will result in higher properties in the finished article. By reference to Table III, it will be seen that even in the soft temper, this alloy (52S–O) is equal in strength to 3S in the hard temper (3S–H). In the quarter-hard temper (52S–$\frac{1}{4}$H), its properties are all definitely superior to those of 3S–H, and it can be formed much more readily than the latter material.

While it is one of the newer alloys, it has already supplanted some of the older alloys for many purposes. It is used for camera cases, for parts of telephone equipment, for panels in buses, trucks and railway cars. It is also coming into use in aircraft for those parts in which the higher properties of 17S–T and 24S–T are not required. Its excellent resistance to salt air, combined with its good mechanical properties, makes it desirable for use on shipboard and for other marine applications.

Heat-Treatable Wrought Alloys

The heat-treatable wrought alloys make available structural materials having mechanical properties comparable with those of structural steel and yet having approximately one-third the weight.

The oldest of these alloys, 17S, substantially the same as the original duralumin, is still one of the most widely used. However, some of the newer alloys of the duralumin type are gradually displacing it in certain fields.

In aircraft, the yield strength of 24S–T (about 25 per cent higher than that of 17S–T), makes possible more efficient design with correspondingly improved performance. In the newer military and transport planes, alloy 24S–T has largely displaced 17S–T. This alloy, 24S–T, with its yield strength approximately 50 per cent greater than that of 17S–T, is the material that has been used in the large shovel dippers, in bridge construction, in heavy-duty trucks, and in similar fields.

Alloy 53S is available in several different tempers depending upon the type of product and the mechanical requirements of the service it is to perform. Extruded sections, which are extensively used, are frequently used without any heat-treatment or with a partial heat-treatment (53S–T5), which produces a distinct improvement in the yield strength. If the material is to be subjected to severe forming before it is used, it is usually ordered as 53S–W and the fabricated article may then be subjected to the precipitation heat-treatment to develop the properties of 53S–T. Where the maximum properties of the alloy are required, the material may be ordered in the fully heat-treated temper. In this temper a considerable amount of forming may be done although more liberal radii are required for bends than in the softer tempers.

The alloy 51S, one of the older alloys developed in this country, has many excellent qualities. It is continuing to hold its position in the face of the competition with the increasing number of newer alloys. In the quenched temper (51S-W), it has good forming qualities and mechanical properties adequate for many purposes. Like 53S-W, it may be artificially aged even after forming to develop the properties of the fully heat-treated temper. After precipitation heat-treatment to form 51S-T, its yield strength is somewhat higher than that of 17S-T, although its tensile strength is about 20 per cent lower. The cost of manufacture of 51S alloy in its various forms is less than that of many of the other heat-treatable alloys, which fact, together with its good properties, contributes to its choice for many applications for which its strength is adequate.

The Free-Cutting Alloy 11S

The newest member of the family of heat-treatable alloys to be exploited commercially is 11S. This alloy has been developed to provide a light strong material for machining in automatic screw machines. Its use is extending rapidly in a number of fields because of complete success which has attended the efforts to provide a truly free-cutting aluminum alloy.

While the extrusion process has already been mentioned, its importance in relation to the problems of the designer warrants a fuller discussion. Many of the wrought alloys of aluminum are available in the form of extruded shapes. The extrusion process removes the limitation that the metal must be disposed about the neutral axis in such a manner that the resulting shape can be rolled. The designer is no longer confined to the standard structural shapes which are a compromise between engineering efficiency and the restrictions of the rolling process.

The use of extruded sections, designed for the particular purpose they are to serve, has made possible even greater savings in weight than would result from the direct substitution of aluminum alloys for heavier materials, section for section. Reductions in erecting and fabricating costs have also been realized in many cases, because the section has been designed with this object in view. This applies to both pressure extrusion through a die and impact extrusion.

The concluding installment of this two-part article will appear in the April number.

QUESTION AND COMMENT

Variable Intermittent Motion

W. M. HALLIDAY
Baildon, Yorkshire, England

● A timing mechanism was required to produce an intermittent rotative motion of the driven shaft running in the same direction as the continuously rotating drive shaft. The amplitude and timing of the intermittently driven shaft was to be adjustable within certain limits and the moving parts were to be totally inclosed. A compact design of such a mechanism is shown in the accompanying illustration.

The drive to the device is through the shaft *A*. Crankplate *B* is keyed to the shaft *A* and is provided with a radial slot into which is fitted an adjustable crankblock *C*. The block is retained in any desired radial position by lock screw at rear of crankplate *B*. Graduations are marked on one face of plate *B* for setting the throw of crankblock *C*, thereby controlling the degree of movement imparted to the driven end of the mechanism. The block *C* has a projecting pin or stud upon which is located a loosely fitted driving shoe *F*. This shoe engages the slot in the adjacent face of the segment or oscillating arm *G*, swinging on stud *H* permanently fixed into the wall of the body *Q*.

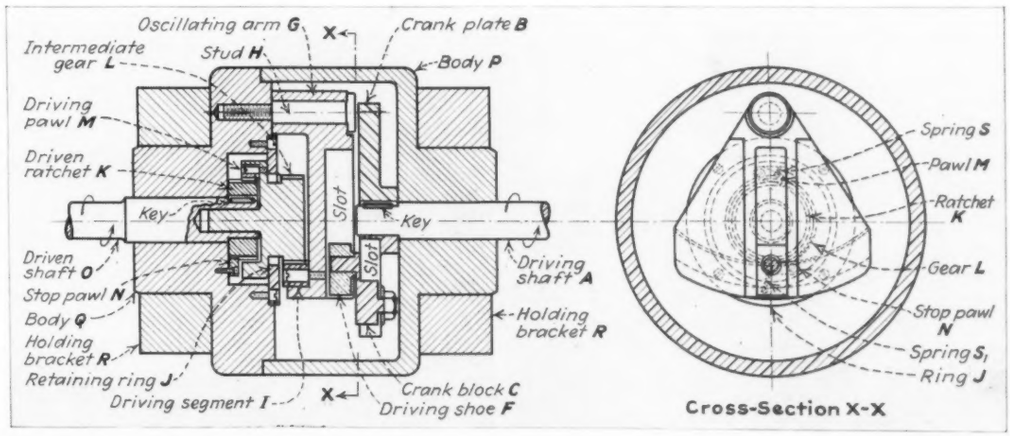
Fastened to the reverse face of arm *G* is an internally cut gear sector *I*. The gear sector meshes with intermediate gear *L*, recessed at one side, into which is fitted driving pawl *M*. This pawl engages the teeth of the driven ratchet wheel *K*, is fastened to the driven shaft *O*, and retained by ring *J*. The ratchet end of the driven shaft is reamed to pilot the intermediate gear *L*.

To prevent reverse movement of the driven parts, a small stop pawl *N* is provided, pivoting freely upon a stationary stud in body *Q*. The rear side of pawl *N* has a drilled hole accommodating a small compression spring S_1 to maintain constant mesh with ratchet *K*. By this means, ratchet *K* is held stationary during the reverse movement of device, while pawl *N* ratchets throughout the forward or work-performing stroke of mechanism. The stationary main body, held in bracket *R-R* is made in two parts as indicated at *P* and *Q*, these being fastened together.

In operation, when the driving shaft *A* is started, motion is at once imparted to the oscillating arm *G* and thus transmitted by the internal gear sector *I* and gear *L* to the driven ratchet *K* on driven shaft *O*.

Because of the pendulum action of the oscillating arm *G*, a slow forward speed and a quick reversal speed is obtained as the center distance of fixed stud *H* to the center of pin on block *C* undergoes a regular increase throughout one half revolution of crankplate *B*, and diminishes during the second half. Thus the driving shoe *F* travels through an orbit farthest removed from the center of stud *H* during the period of slow forward speed.

This oscillating motion then is conveyed from the arm *G* to the driven ratchet *K* causing the ratchet to be actuated in one direction only as permitted by the driving pawl *M*. Stop pawl *N* and its compression spring prevents any possibility of the driven ratchet *K* and its shaft *O* from reversing through friction of the parts.



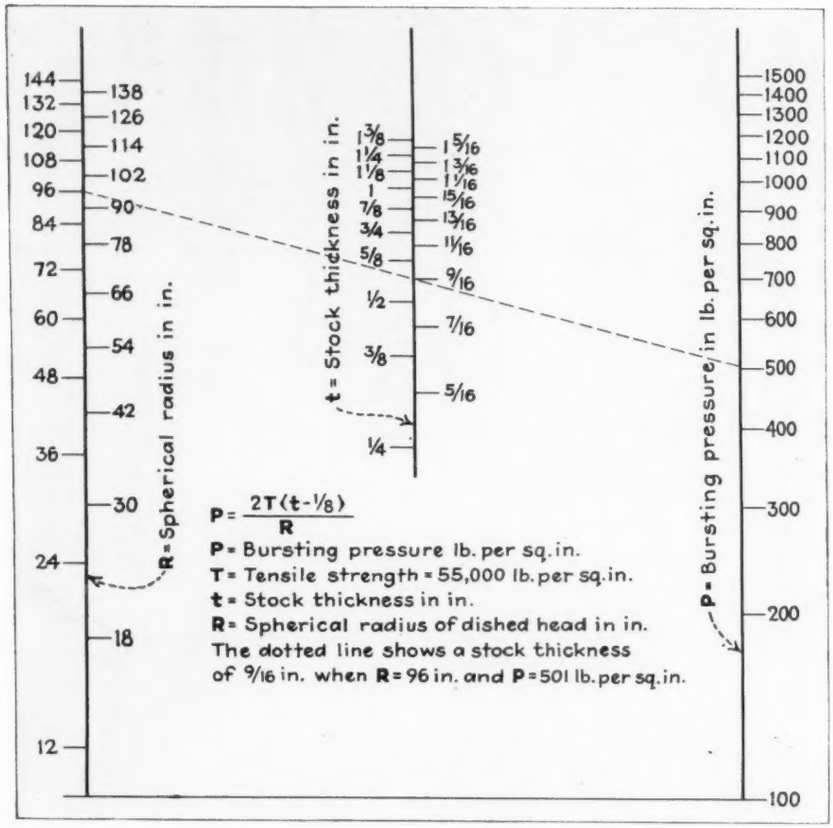
The intermittently driven shaft rotates in the same direction as continuously revolving drive shaft. A ratchet arrangement prevents reversal of drive

Effect of Specimen Length on Test Data

To the Editor:

● The article under the above title by Victor Tatarinoff on page 347 in the September number brings out the little recognized fact that under certain conditions, length has an effect on the torsional resistance of a flat bar. The author, however, fails to mention the conditions under which that effect applies; nor does he mention other factors that are of equal importance in determining the torsional resistance of such a bar.

To take the simplest example, a thin flat spring or piece of celluloid is twisted so that no restraint is imposed other than pressure at right angles to the surface at each of the four corners. Every diagonal element of the surface is bent, although all transverse elements and the central longitudinal element remain straight. Since the diagonal elements are bent, they are subject to bending moment even when the applied torque is not sufficient to cause measurable twisting in the bar. Bending moment is accompanied by



Thicknesses of "plus" and "minus" dished heads of tanks can be read directly from nomogram when bursting pressure P and spherical radius R of head are known

tension and compression in opposite faces of the bar.

The resultant stress is much greater than either the shear or the tensile stress alone, and failure must occur not by shear, but by what is known to reinforced concrete designers as diagonal tension. Therefore, a flat bar must fail diagonally in torsion. Length or the ratio of length to width has no effect on torsional resistance, since stresses are uniform throughout the length of the bar.

If the torque is applied through rigid clamps extending along the ends of the bar, as is more commonly done in practice, additional conditions are imposed that are similar to those imposed on a simple beam by fixing the ends. Longitudinal bending moments are introduced, varying from a maximum at the edges to zero at the center of the bar. These bending moments bring about a realignment of stresses with the result that in comparatively short bars, failure must occur under less torque than is necessary to cause failure when torque is applied without clamping.

Further, stresses are no longer uniform throughout the length of the bar and consequently, length has an effect on the torsional resistance. The longer the bars the less the variation of stresses, and the more nearly the torsional resistance approaches that of the unclamped bar. The relationship between torsional resistance and the ratio of length to width is not an exponential one, however, as would be inferred from the straight line logarithmic diagram given by Mr. Tatarinoff.

The practice of using formulas blindly without regard to the limiting assumptions is too prevalent. The shear formula for torsional resistance can be rigorously applied only to solid or hollow circular shafts. It takes into account the shearing strength of the material only. In sections that are not circular, the unit tensile strength, the transverse elastic modulus, and the tension modulus of the particular material all have an influence on torsional resistance. The unit shear strength plays a minor part.

This situation is similar to the use of the Kutter formula for the flow of water in open channels. Engineers go on adding decimals to the values of "n" to fit observations of flow in different channels while neglecting the fact that the fluidity of water may vary 30 per cent or more between summer and winter temperature.

Formulas are comfortable sedatives to thought. However, intelligent use of these relations requires knowledge of their limitations.

—Leo G. Hall
Blood-Brothers Machine Company

Errata

● In the solution of the optical extensometer as submitted by L. G. Hall on page 22 of the January number, the divisor, 6 cos θ, of the equation, should be 4 cos θ. Therefore, $X = (175 - 4.969)/4 \cos \theta$ is the corrected formula.

Chart for Tank Heads

C. EDWIN LEWIS
Dowmingtown Iron Works

● Calculating the required stock thickness of a spherical head for pressure tanks, when estimating costs, usually takes more time than is justifiable. The accompanying alignment chart not only provides a quick estimate of plate thickness for a given head radius and pressure, but also affords a check on design calculations.

The chart refers only to so-called "standard heads," having a spherical radius approximately the same as the outside diameter of the tank. For "plus" heads (pressure on concave surface of head) bursting pressure $P = P_1 f$. For "minus" heads (pressure on convex surface of head) $P = 1.667\ P_1 f$. Working pressures P_1 are not included in the nomogram, but can be calculated from the following equation when $f = 4$ to 5.5 = factor of safety.

$$P_1 = \frac{2T\left(t - \frac{1}{8}\right)}{f\,R}$$

This design allows a smaller factor of safety than is specified by the A.S.M.E. Code for Unfired Pressure Vessels, and is intended only for general tank construction.

Solution of Optical Extensometer Problem

To the Editor:

● In the diagram under the title "Dimensions Wanted for Optical Extensometer," by W. W. Mohr on page 469 of the December issue, it is assumed that the two mirrors are exactly parallel at the beginning of the test. The position of the beam coming to the first mirror will then be parallel to the beam leaving the second mirror. It is also assumed that the deformation is the same on both sides of the test specimen.

As in the example shown in the accompanying diagram, c = linear elon-

gation of the specimen; E = linear displacement of beam on drum; α = angular displacement of mirrors; and θ = angle between beam and mirror at start.

The angle of incidence being equal to the angle of reflection the points A, B and C are so noted. Therefore:

$$AB = 2.484/\sin\theta$$
$$AC = AB \text{ (approximately)}$$
$$BD = AB \sin(180 - 2\theta)$$
$$CD = AB \sin[(180 - 2\theta) + 2\alpha]$$
$$BC = CD - BD = \frac{-2.484 \cos 2\theta \sin 2\alpha}{\sin\theta}$$
$$KH = X \sin 4\alpha$$
$$HG = BC$$
$$E = KH + HG$$
$$E = X \sin 4\alpha - \frac{2.484 \cos 2\theta \sin 2\alpha}{\sin\theta}$$

Since angle α is very small, sin 2 α will nearly equal 2 sin α and similarly sin 4 α will nearly equal 4 sin α. The desired magnification being 1,000, the equation for E/e will be:

$$\frac{E}{e} = 1{,}000 = \frac{4X}{0.175} - \frac{4.968}{0.175}\,\frac{\text{Cos } 2\theta}{\text{Sin } \theta}$$
$$X = 43.75 - 1.242\,\frac{\text{Cos } 2\theta}{\text{Sin } \theta}$$

From which the value of X can be determined when θ, the angle between the beam and mirror, is known.

If the angle θ = 60 deg., X would then be 43.75 — 0.717 = 43.033 in. in this example.

—NORMAN M. WICKSTRAND, *Bristol, Conn.*

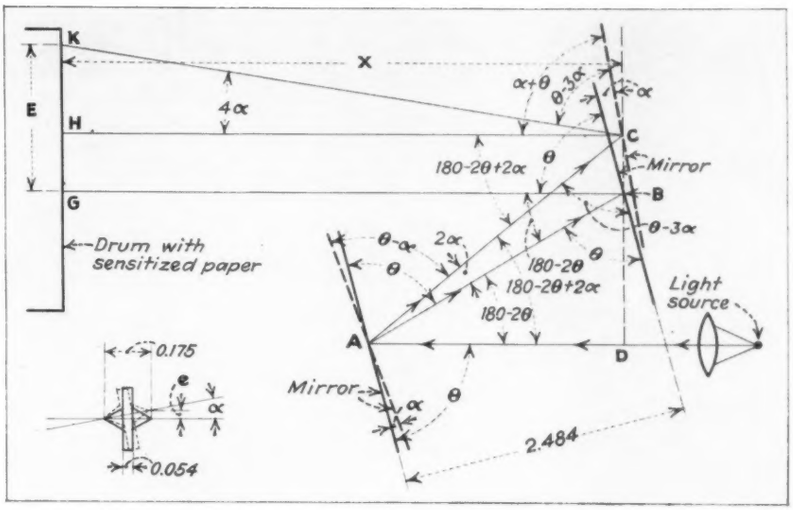


Diagram of mirror positions for optical extensometer with magnification of 1,000 times

Length of Material in Cylindrical Rolls

J. STILES BEGGS
Research Laboratories
Eastman Kodak Company

● When it is desired to know the approximate length of material in a roll of paper, strip metal, or other material, the accompanying alignment chart will give this information quickly. If D is the outside diameter of the roll, d the core diameter, and t the thickness of the stock, all expressed in inches, the length L in feet is,

$$L = \frac{\pi}{48\,t}(D^2 - d^2)$$

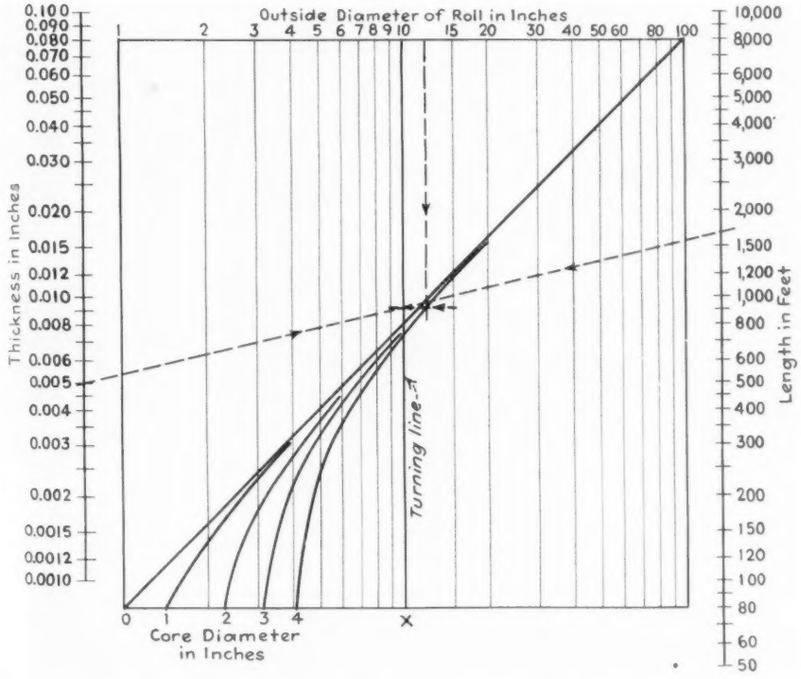
This relation applies when the strip thickness is small compared with the core and roll diameters.

Three steps are required in using the chart. First, a vertical line is drawn from the given outside diameter to the core diameter curve. From the point of intersection a horizontal line is drawn intersecting the vertical line X. This intersection is used as a turning point for a line passing through the thickness scale and intersecting the length scale which gives the length of material in the roll.

As shown by the dotted lines on the chart, a roll 12 in. outside diameter on a core 4 in. in diameter, has a length of 1,680 ft. for paper 0.005 in. thick.

If the material is wound loosely the thickness of the material as determined by a micrometer will not give the correct length from the chart. However, a nominal thickness for the material can be determined from the chart if the length is known or measured for one roll. This nominal thickness is the actual thickness of the material plus the clearance between successive layers. For other rolls wound in the same way, the measured thickness can be corrected so as to give the proper length.

For example, if a roll contains 4,000 ft. of paper 0.005 in. thick and has an outside diameter of 20 in., the chart indicates a thickness of 0.0064 in. Hereafter, for this material a nominal thickness of 0.0064 in. is used instead of the actual thickness of 0.005 in. Using the nominal thickness of 0.0064 in., for the example shown on the chart gives a length of 1,300 ft. for a roll 12 in. outside diameter on a core 4 in. diameter.



Alignment chart for obtaining the approximate length of thin material in a roll when outside and core diam. are given

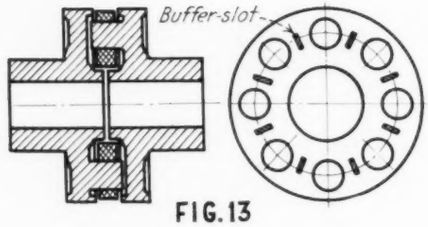


FIG. 13

Typical Designs of

CYRIL DONALDSON
Rochester Mechanics Institute

Second of a series of couplings having leather forms, metal disks or wire cable links to provide the necessary flexibility

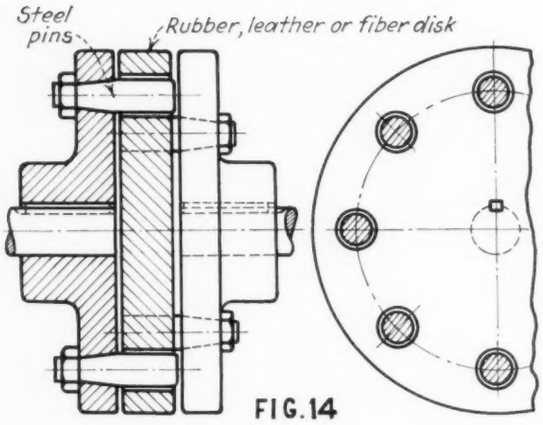


FIG. 14

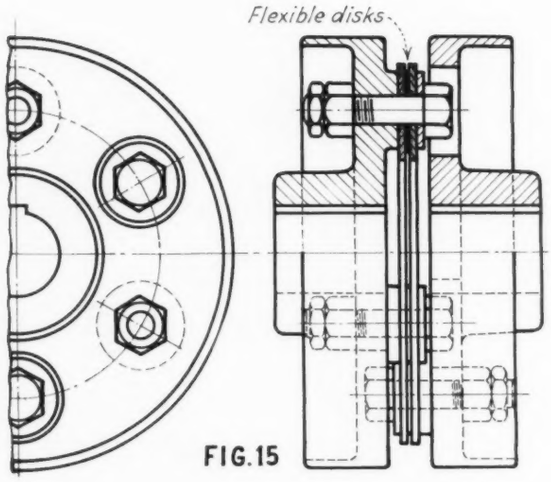


FIG. 15

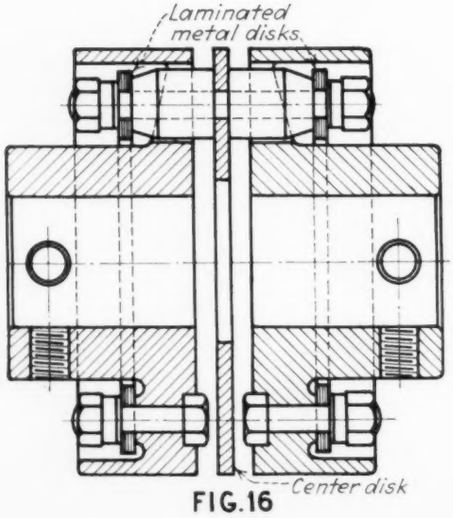


FIG. 16

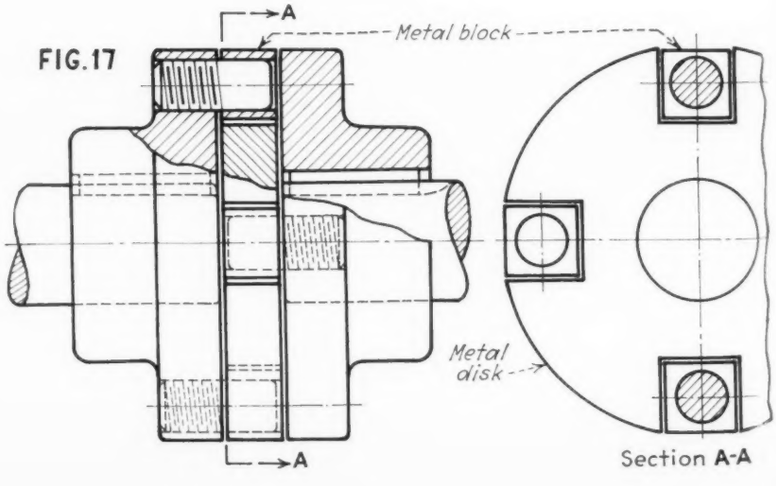


FIG. 17

Fig. 13—In this Brown Engineering Company coupling flexibility is increased by addition of buffer-slots in the laminated leather. These slots also aid in the absorption of shock loads and torsional vibration. Under parallel misalignment or shock loads, buffer slots will close over their entire width, but under angular misalignment buffer slots will close only on one side.

Fig. 14—Flexibility is provided by resilience of a rubber, leather, or fiber disk in this W. A. Jones Foundry & Machine Company coupling. Degree of flexibility is limited to clearance between pins and holes in the disk plus the resilience of the disk. Has good shock absorbing properties, allows for end play and needs no lubrication.

Fig. 15—A coupling made by Aldrich Pump Company, similar to Fig. 14, except bolts are used instead of pins. This coupling permits only slight endwise movement of the shaft and allows machines to be temporarily disconnected without disturbing the flanges. Driving and driven members are flanged for protection against projecting bolts.

Fig. 16—Laminated metal disks are used in this coupling made by Thomas Flexible Coupling Company. The disks are bolted to each flange and connected to each other by means of pins supported by a steel center disk. The spring action of the center ring allows torsional flexibility and the two side rings compensate for angular and offset misalignment. This type of coupling provides a positive drive in either direction without setting up backlash. No lubrication is required.

Fig. 17—A design made by Palmer-Bee Company for heavy torques. Each flange carries two studs upon which are mounted square metal blocks. The blocks slide in the slots of the center metal disk.

Flexible Couplings—II

Fig. 18—In this Charles Bond Company coupling a leather disk floats between two identical flanges. Drive is through four laminated leather lugs cemented and riveted to the leather disk. Compensates for misalignment in all directions and sets up no end thrusts. The flanges are made of cast iron and the driving lug slots are cored.

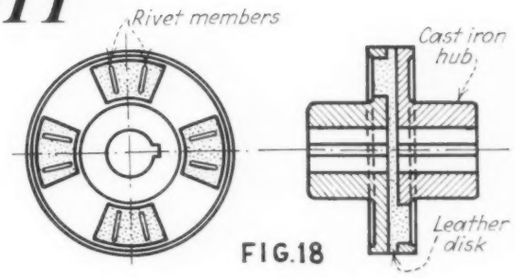


FIG. 18

Fig. 19—The principle of the T. B. Wood & Sons Company coupling is the same as Fig. 18, but the driving lugs are cast integrally with the metal flanges. The laminated leather disk is punched out to accommodate the metal driving lugs of each flange. This coupling has flexibility in all directions and does not require lubrication.

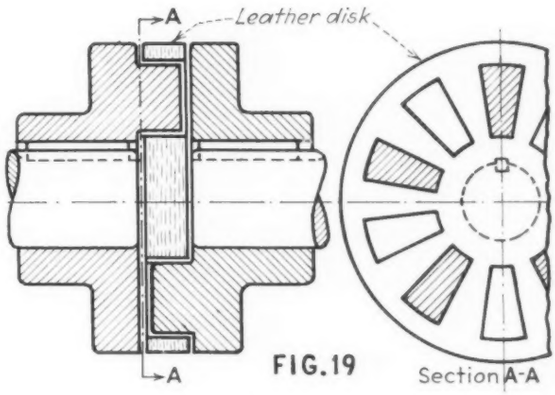


FIG. 19

Fig. 20—Another design made by Charles Bond Company. The flanges have square recesses into which a built-up leather cube fits. Endwise movement is prevented by through bolts set at right angles. The coupling operates quietly and is used where low torque loads are to be transmitted. Die-castings can be used for the flanges.

Fig. 21—Similar to Fig. 20, being quiet in operation and used for low torques. This is also a design of Charles Bond Company. The floating member is made of laminated leather and is shaped like a cross. The ends of the intermediate member engage the two cored slots of each flange. The coupling will withstand a limited amount of end play.

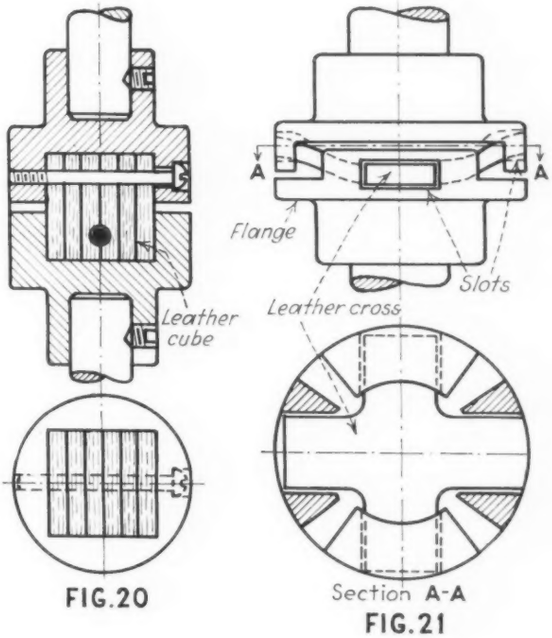


FIG. 20
FIG. 21

Fig. 22—Pins mounted in flanges are connected by leather, canvas, or rubber bands. Coupling is used for temporary connections where large torques are transmitted, such as the driving of dynamometers by test engines. Allows for a large amount of flexibility in all directions, absorbs shocks but requires frequent inspection. Machines can be quickly disconnected, especially when belt fasteners are used on the bands. Driven member lags behind driver when under load.

Fig. 23—This Bruce-Macbeth Engine Company coupling is similar to that of Fig. 22, except that six endless wire cable links are used, made of plow-steel wire rope. The links engage small metal spools mounted on eccentric bushings. By turning these bushings the links are adjusted to the proper tension. The load is transmitted from one flange to the other by direct pull on the cable links. This type of coupling is used for severe service.

FIG. 23

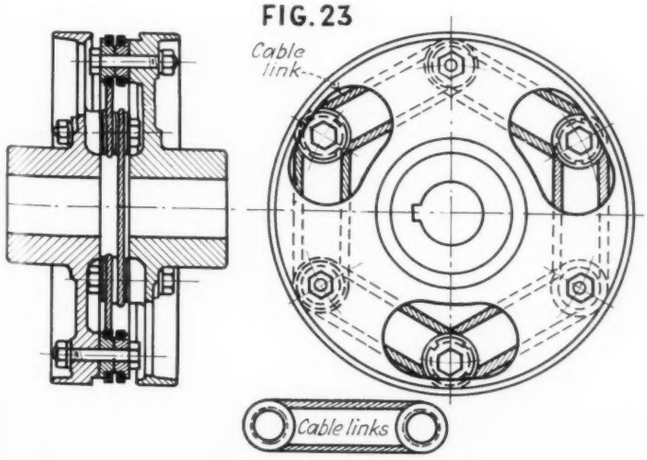
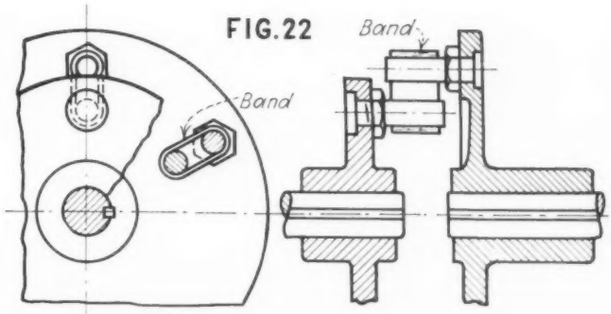
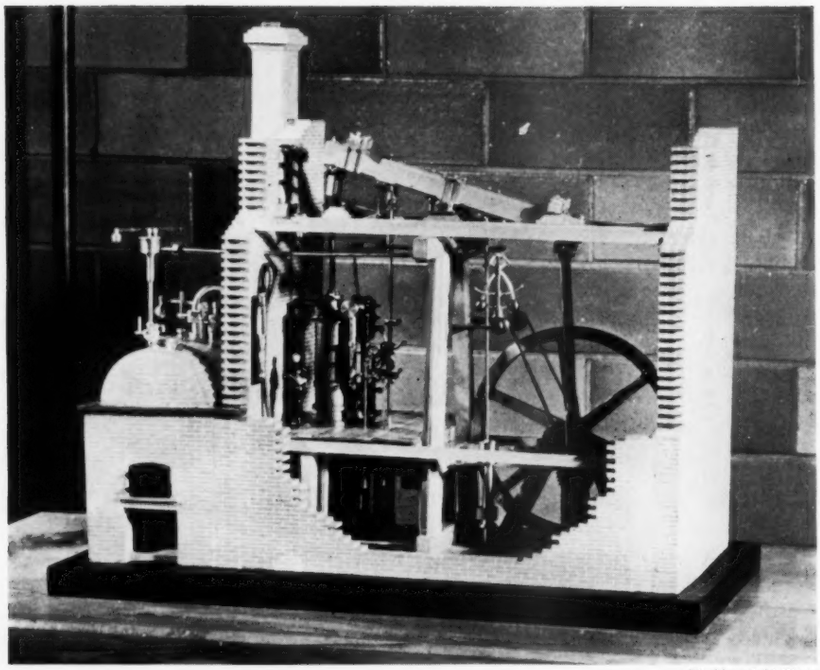


FIG. 22





W. Mansfield White

Working model of the original Watt engine constructed in the shops at Lehigh University. The model is said to be the only one in existence that runs under its own power

NEWS

Birth of Watt Celebrated

Engineers talk on life and work of James Watt and discuss problems of the college graduate in industry

MORE than 1,000 men prominent in education and industry attended the bicentennial celebration of the birth of James Watt held at the Franklin Institute in Philadelphia and Lehigh University in Bethlehem, Pa., January 19-21 inclusive. The program was arranged under the joint auspices of Lehigh University, The Franklin Institute of the State of Pennsylvania, the American Society of Mechanical Engineers and the Newcomen Society of England.

The first meetings of the celebration were held at the Franklin Institute on Sunday January 19. A feature was the opening address by H. W. Dickinson speaking at the Science Museum in South Kensington, London, and broadcast internationally. After the broadcast, models of the Newcomen and Watt engines on display at the Franklin Institute were inspected.

The main part of the program consisted of a series of addresses and discussion on Monday, January 20, at Lehigh University. In addition to addresses related to the life and work of James Watt, a discussion was held on the problem of "The College Graduate in Industry." Among the personnel directors who participated in this discusion were O. W. Eshbach of the American Telephone & Telegraph Company; C. S. Coler of the Westinghouse Electric and Manufacturing Company; N. J. Hittinger of the Bethlehem Steel Corporation; M. M. Boring of the General Electric Company C. W. Beese, industrial engineer of the Armstrong Cork Company; H. B. Bergen, director of industrial relations of the Proctor & Gamble Company; and E. R. Morgan, director of placement at Lehigh University, also presented discussions on this important subject.

The afternoon sessions included papers on the life and work of James Watt, whose improvement of the steam engine lead to the train of events which introduced what is known as the industrial revolution. Also, at the afternoon session Moreland King professor of electrical engineering at Lafayette College, and Frank B. Bell president of Edgewater Steel Company each gave an address on "Looking to the Future" as viewed by the Educator and Industrialist respectively.

Following a dinner, the evening meeting was held in the Packard Auditorium. William L. Batt, president of The American Society of Mechanical Engineers, spoke on "Watt—Symbol of the Industrial Age," and W. C. Dickerman on "Some Problems of a College President."

The concluding sessions of the celebration were held at the Franklin Institute on Tuesday afternoon concluding with a formal dinner at the Bellevue-Stratford Hotel in Philadelphia.

In both the afternoon and the evening sessions, talks by prominent engineers and executives brought out the influence of steam on civilization as well as its importance to railroads and electrical utilities.

A. S. M. E. Sponsors Session on Industrial Design

Illustrated talks, demonstrations and exhibits will show how to improve the appearance of mechanical products

MEMBERS and guests of the Metropolitan Section of the American Society of Mechanical Engineers will have an opportunity to see how industrial designers and color experts apply their methods to mechanical products on February 20 in a meeting to be held at the Engineering Societies Building. In a session on "Improving the Appearance of Mechanical Products," three speakers will show how both color and redesign have been applied to increase sales, improve appearance and reduce costs in engineering products ranging from locomotives, automobiles, machine tools, and business machines to vacuum cleaners and alarm clocks.

The chairman of the meeting, R. W. Carson, assistant editor of *Product Engineering*, will show in a series of actual examples how color and indus-

trial design have influenced the sales success in a number of products placed on the market during 1935. These examples will show "before and after" contrasts, as well as data on the effect of the redesign on costs, sales and profits.

Following these examples of profitable applications, Howard Ketcham, well known to *Product Engineering* readers for his series of color articles, will show how colors are selected for industrial products, and will also present the results of several recent surveys showing current color preferences for different price classes and different geographical divisions. Fundamentals that a mechanical engineer needs to understand in applying color will be demonstrated. From actual applications a number of examples will be drawn showing how color influences the performance as well as the appearance of machines.

In the presentation of the practical side of industrial design Henry Dreyfuss, one of the outstanding industrial designers, will show how the product designer cooperates with the engineer, using as examples a number of recently developed and successful designs. Methods used by the industrial designer will be demonstrated throughout the entire procedure from the idea to the sketches, the models and blueprints, through production to the finished machine. Work now being done on a new train for the New York Central Railroad will be featured.

Company Name Changes

EFFECTIVE the first of this year the following three companies have adopted new names.

Whitney Manufacturing Company is now the Whitney Chain & Manufacturing Company.

Plaskon Company is the new name for the organization formerly known as Toledo Synthetic Products, Inc.

W. M. Chace Company is the name now adopted by the bimetal manufacturers formerly known as W. M. Chace Valve Company.

New Bearing Material Coming

A PROMINENT Detroit bearing manufacturer will shortly announce a new material having more of the characteristics of babbitt metal than of either cadmium or copper-lead. While it is too early to announce the material as yet, preliminary experiments have shown it to be capable of higher unit load than the conventional automotive babbitts without the objectionable crankshaft cutting often experienced with copper lead. The new alloy is said to have no adverse reactions to the fatty acids which have proven to be the main drawback for the cadmium bearing. An independent laboratory is just completing the final test on the new material.

MEETINGS

American Institute of Mining & Metallurgical Engineers — Annual meeting, Institute of Metals Division, Engineering Societies Bldg., New York, N. Y. Feb. 17-21. A. B. Parsons, secretary, 29 West 39th St., New York, N. Y.

American Society for Testing Materials—Regional meeting, William Penn Hotel, Pittsburgh, Pa., Mar. 4. R. E. Hess, assistant secretary, 260 S. Broad St., Philadelphia, Pa.

American Society for Testing Materials—Annual meeting, Atlantic City, N. J., June 29-July 3. R. E. Hess, assistant secretary, 260 S. Broad St., Philadelphia, Pa.

EXHIBITIONS

Packaging Exposition — Hotel Pennsylvania, New York, N. Y., March 3-6. American Management Association, 232 Madison Ave., New York, N. Y.

National Oil Burner Show—Convention Hall, Detroit, Mich. April 14-18. C. F. Curtin, 30 Rockefeller Plaza, New York, N. Y.

Crane Concentrates Research Activity

ALL RESEARCH and development work in the Crane Company, formerly done in various departments and subsidiaries, has been concentrated in the Chicago Works. A new division entirely divorced from routine production has been formed and will be responsible directly to the president. M. W. Link will head up the new division, with the title director of research.

Announcement was also made of the creation of a new Product Engineering Department operating under the vice-president of manufacture. This group will be headed by A. M. Houser, Jr., and will be responsible for placing new developments into production.

Do You Know That—

- Small wire forms come from one automatic machine so fast that gravity must be assisted by an air blast in order to get the work away from the machine.
- Endurance strength of galvanized steel wire is increased more than 100 per cent by removing 10 per cent of the wire diameter.
- Austenitic cast iron can now be obtained with expansion characteristics ranging from those of aluminum to those of plain cast iron or steel.
- Because round belts slip readily when overloaded, they are becoming so popular as overload releases that one manufacturer is coming out with a greatly expanded line of endless round belts.

For more details write the Editor

McGraw-Hill Changes Officers

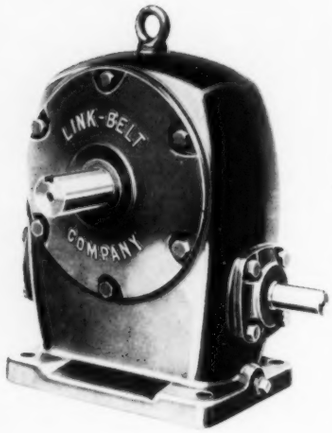
AT A meeting of the Board of Directors of the McGraw-Hill Publishing Company, Inc., held on December 27, 1935, James H. McGraw, the founder of the company and its head for more than fifty years, resigned as Chairman of the Board and was elected Honorary Chairman. He will remain as a member of the Board.

James H. McGraw, Jr., who has been connected with the company for the past twenty years, was elected Chairman of the Board. He has served as Treasurer and was Executive Vice-President and Vice-Chairman of the Board at the time of his election.

Malcolm Muir, President of the company since 1928, continues in that capacity.

JAMES H. McGRAW
Honorary Chairman
McGraw-Hill Publishing Company

New Materials and Parts



Worm Gear Reducer

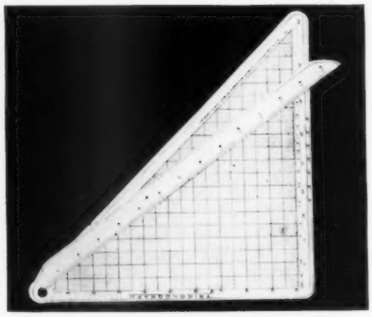
Has either single or double reduction in horizontal or vertical types. Wormwheel is made of chilled phosphor bronze. The worm threads are cut directly on a forged alloy steel shaft carburized and heat-treated. Shafts are mounted on tapered roller bearings having automatic lubrication. The output shaft can be located either above or below the worm. The double-reduction type has a unitized primary reduction attached to the final-reduction housing. Housings are made of gray iron and are dust-proof. Link-Belt Co., 307 N. Michigan Ave., Chicago, Ill.

Hi-Steel

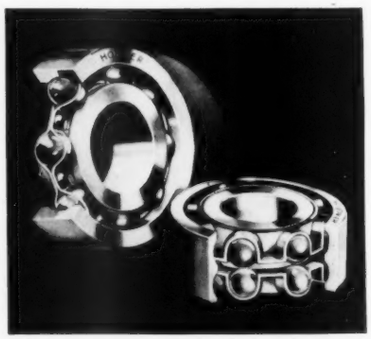
Low alloy, high-strength steel has an ultimate tensile strength of 70,000 lb. per sq.in. Minimum yield point for gages lighter than $\frac{1}{4}$ in. is 60,000 lb. and for heavier gages, 55,000 lb. per sq.in. It is said that this steel can be bent, formed, stamped, seamed, and welded, and is highly resistant to corrosion. Available in sheets, strips, plates, bars and structurals. Inland Steel Co., First National Bank Bldg., Chicago, Ill.

Trigonograph

Is used as a time saver for checking calculations or where results of two or three places are satisfactory. Composed of a 9 in., 45 deg. right triangular base and a pivoted hypotenuse, both of which are decimally divided. By moving the hypotenuse through an angle the ratio between the sides of a right triangle are graphically indicated. An auxiliary scale is also provided for the solution of oblique triangles. Available in models made of cardboard or of opaque non-warping celluloid. A 16-page instruction book is included. Chemical Rubber Publishing Co., 1900 West 112th St., Cleveland, Ohio.



Ball Bearings



New sizes of radial and angular contact types, 75 in number, have been added. These sizes embrace both single and double-row types made to S.A.E. and U. S. Navy specifications covering design, tolerances, materials and other factors. Single-row bearings include all sizes from No. 200 to 222 in the light series, No. 301 to 318 in the medium sizes and No. 404 to 410 in the heavy series. Double-row bearings are in sizes of No. 5202 to 5210 in the light series and No. 5302 to 5312 in the medium sizes. Both single and double-row types are also available with one or two lubraseal plates. A number of special sizes have been added for specific applications in passenger cars, trucks and tractors. The Hoover Steel Ball Co., Ann Arbor, Mich.

Bright Zinc Plating

The "Mazic" process for zinc plating permits heavy deposits from a cyanide solution said to produce a bright rather than a white finish. Also said to deposit coatings comparable to cadmium finishes in still tank, semi-automatic and full automatic plating systems. It is claimed that with the use of Mazic anodes in an unlined steel tank, the solutions are easy to maintain, and that the high cathode current density makes this a high speed plating bath. Hanson-Van Winkle-Munning Co., Matawan, N. J.

Hydraulic Pump

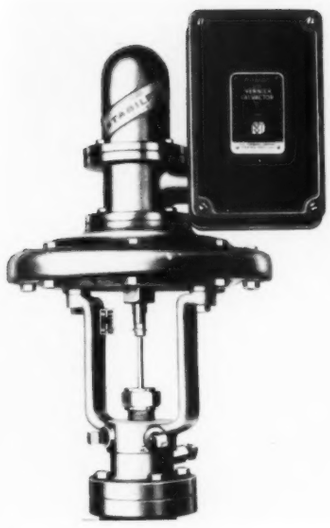
Unit consists of a variable displacement pump having two adjustable feeds and a constant displacement pump that provides rapid traverse. Both pumps are incorporated in a single housing and driven by a single shaft. The variable displacement pump is of the multiple piston type, which gives a positive uniform feed. Two pre-set rates of feed are provided, both of which are adjustable throughout their entire range.

The constant displacement pump has a roller that revolves in positive contact with the rotor for smooth operation thereby eliminating sliding blades and gears. Both piston and constant

displacement pumps are said to give quiet operation. The two revolving parts are mounted on ball bearings. Sundstrand Machine Tool Co., Rockford, Ill.

Vernier Valvactor

For air-operated control valves, to eliminate valve sticking and to assure hair-line valve positioning. In air-operated control instruments of throttling type, the device is said to control



valve position regardless of friction or hysteresis. It is further claimed that the air from the control instrument need change as little as ½ in. of water to cause a corrective positioning of the valve and force the stem to take a position within 0.001 in. of the previous position. The unit is mounted in a cast aluminum weatherproof case. Also available in gas-tight construction. The Foxboro Co., Foxboro, Mass.

Thiokol Molding Material

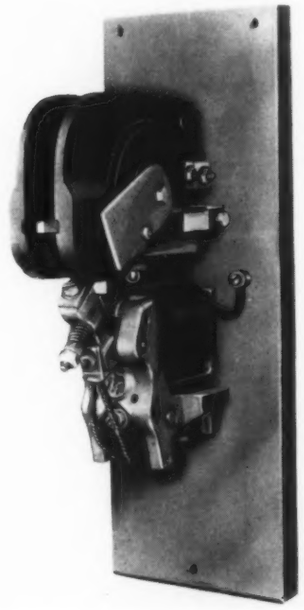
A synthetic rubber compound which can be molded much the same as phenols or ureas and to cure almost as fast. In its molded state the compound is claimed to be fully vulcanized and to be unaffected by gasoline, oil or exposure to air and sun. Used for industrial applications where natural rubber is found to be inadequate. Thiokol Corp., Yardville, Trenton, N. J.

Rotary Geared Pump

Has helical gears for high speed and quietness. Renewable bronze bearings permit ordinary motor speeds. Used for clean liquids such as oil, oil emulsion or soda solution. Can run in either direction and is furnished with or without automatic relief valve. Pumps are said to raise oil on the suction side from 25 to 35 ft. at speeds more than 1,000 r.p.m. Three sizes are available for ⅜, ½ or ¾ in. pipe size connections and a maximum pressure of 30, 40 and 50 lb. respectively. Brown & Sharpe Mfg. Co., Providence, R. I.

Line-Arc Magnetic Contactor

Is used on mill and crane machinery and other d.c. applications. The arc, formed as the circuit is opened, is said



to be ruptured harmlessly in a narrow path centered between, but not touching, the arc shields. Contactors are made of aluminum alloy and rolled steel, permitting a quick response and speedy operation because of the light weight and low inertia. Available up to and including 800 amp. capacity. The Electric Controller & Mfg. Co., 2700 E. 79th St., Cleveland, Ohio.

Electrode for Welding Stainless Steel

Because of its higher chrome content, this new electrode can be used for welding stainless steel containing 25 per cent chromium and 12 per cent nickel. The weld metal is said to have high corrosion-resistance and to give as high a tensile strength and ductility as possessed by 25-12 alloy steels. Welds are also said to be free from oxides and nitrides and porosity. Said to resist stresses to 95,000 lb. per sq.in. The new electrode is known as "Stainweld B." The Lincoln Electric Co., Cleveland, Ohio.

Cord Belt

New rubber-duck, mahogany-colored belt has square-cut edges. Because the cords are embedded in rubber and the edges square-cut, the belt is said to withstand rubbing against shifter forks. The laminated and cut edge is said to permit the use of a single ply of duck on each side instead of several plies in envelope form. A special duck, requiring but one cover joint, has been developed to eliminate a number of seams as in a bias-cut duck cover. Because of the method of manufacture it is claimed that a smooth joint can be made for endless belts. Belts are available to 24 in. wide and in five or more plies. United States Rubber Products, Inc., 1790 Broadway, New York, N. Y.

Water-Resistant Molding Material

A new material, known as Durez 75 Series is used for molded parts requiring waterproof characteristics. Parts made of this material are said not to be affected even by complete and continual immersion in water. Also said to withstand alcohol solutions. Can be used for float balls in tanks, and containers for semi-liquids or creams having a large water content. General Plastics, Inc., North Tonawanda, N. Y.

Inclosed D.C. Motor

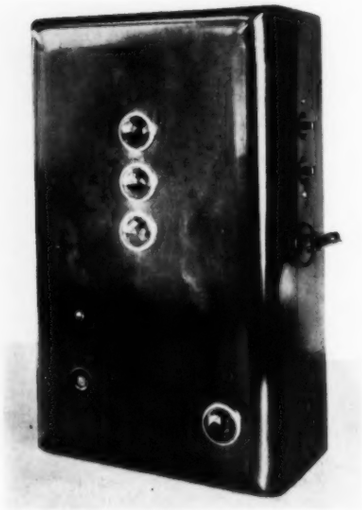
Totally inclosed d.c. motor is fan cooled, having an external fan and shroud on the commutator end. Removal of but one cover on bracket permits accessibility to brushes. Air ducts are cast into bearing bracket

and outer air shroud. By placing the blower on the commutator end the motor can be mounted against a gear case or other wall. It is not necessary to disturb the shroud in dismantling the motor. Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.

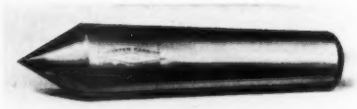
Rustproofing With Cromodine

A rustproofing process known as "Cromodizing" is used as a pre-treatment for metal parts to protect them against rust and corrosion. The parts are dipped in or sprayed with a solution of cromodine for one minute. Parts are then rinsed, dried and painted as desired. The treated surface is said to give the metal a smooth surface which is receptive to paint, also to triple the life of steel parts so treated. It is claimed that the finish cannot be broken when the metal is bent or distorted and that no unusual equipment for the process is required. American Chemical Paint Co., Ambler, Pa.

Alternatector



Provides means for alternately operating electric circuits in installations where duplicate devices are used. The alternator provides a master switch that does not carry current when operated and therefore does not arc. Is intermediate to primary and secondary switches, connected by wires only. Pilot lights indicate which circuit is in operation. Red pilot light and ringing gong signal failure. Available in several combinations. Is used in industrial service for dual oil burners, dual solenoid valves and motor operated valves. Is 19 in. high, 14 in. wide and 6½ in. deep. Enterprise Foundry Corp., 2002-2008 Nineteenth St., San Francisco, Calif.



Tipped Lathe Center

Because of their widespread use, tungsten carbide tipped centers have been added in regular production by this company. Used for both finishing and roughing operations and especially for hogging where side thrusts are excessive. Said to give long life and minute wear so that truing time is reduced to a minimum. Tungsten Carbide Tool Co., Detroit, Mich.

Oil Pump

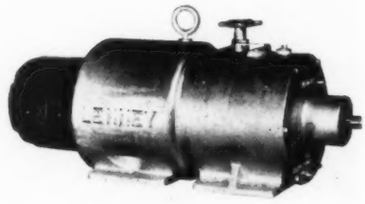
For cutting oils and coolants for all classes of machine tools and for circulating solvents. Is said not to lose its prime, being accomplished by the use of a vacuum dome. Pump is pulley driven with round, flat or V-belts. Has a capacity to 5 gal. per min. at 600 r.p.m. Pipe leads can be taken off at several positions. Hudson Mfg. Co., Oshkosh, Wis.

Seismo-Damper

Type T is used for vibration elimination. Machine to be supported rests on a T-iron separated from a base of two angle irons by natural cork or other vibration isolating material. The connecting bolts in the angle irons pass through enlarged holes in and not touching the resilient material. The bolts are adjusted to produce a pre-compression to suit the machine load. When machine must be supported at several points in a straight line, the T-iron can be extended to form a continuous support between two or more dampers. A stud in the T-iron engages the hole in the machine foot. The Korfund Co., 48-15 Thirty-Second Place, Long Island City, N. Y.

Alnico Magnets

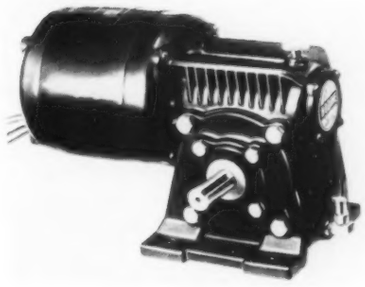
Permanent magnets said to be more powerful than hitherto available are made of a new alloy of aluminum, nickel, cobalt and iron. This alloy contains no carbon and is of the precipitation hardening type. Coercive force is higher and the residual magnetism lower than for other permanent magnet materials. The maximum available energy is also higher, but occurs at a lower flux density. Alnico magnets therefore may be of much smaller volume than conventional magnets, but generally require a larger cross section. Because of the high coercive force, a much shorter length can be used. Alnico magnets are said to be less affected by vibration and high temperature, and less subject to demagnetization. Magnets are cast to shape and finished by grinding. Holes for mounting are cored or soft steel inserts are cast in. Simonds Saw & Steel Co., Lockport, N. Y.



Variable-Speed Transmission

A self-contained mechanical speed-reducing transmission of the over-running clutch type. Used between prime movers and machines where a low variable speed is necessary. Is infinitely adjustable to a fraction of a r.p.m by a handwheel while still or running. Speed is indicated by a pointer on top of housing. All moving parts are mounted on anti-friction bearings or hardened and ground steel bushings. All moving parts run in a bath of oil. Said to be quiet running and to require no attention. The Lenney Machine & Mfg. Co., Warren, Ohio.

Worm Gear Speed Reducer



Heavy-duty reducer for all Bodine N-5 motors. Unit is rated to carry ¼ hp. continuously with a 1/6 hp. motor. Available in gear ratios from 8 up to 48 to 1. Can be mounted with motor shaft horizontal or vertical, or with shaft parallel to or running through the mounting surface. Shaft can be extended in either or in both directions, and a high-speed shaft can be extended from the motor. Gear housing and motor end shield are cast in one piece. Drive shaft is supported on two double-row ball bearings of the sealed type. Bodine Electric Co., 2264 W. Ohio St., Chicago, Ill.

MANUFACTURERS' PUBLICATIONS

Ball Bearing Lubrication — Hoover Steel Ball Co., Ann Arbor, Mich. Bulletin 11x17 in. Lists in tabular form recommended greases for lubricating ball bearings in all sizes. Temperature ranges of below 32 deg. to 250 deg. and speeds of 100 to more than 10,000 r.p.m. are also given. General specifications as to consistency, base and moisture of 9 grease manufacturers are contained in the recommendations.

Bearing—Temperature Relays—General Electric Co., Schenectady, N. Y. Leaflet, GEA-2209. Illustrating type TB-2 relay, of instantaneous, bellows-type with hand-reset for protecting machine bearings against abnormal heating. A detailed line drawing with captions shows the features.

Bronze Bushings and Bar Stock—Johnson Bronze Co., New Castle, Pa. Bulletin No. 360, 40 pages. General catalog, 1936 edition, gives weight and sizes of completely machined bars both cored and solid and lists over 600 sizes of general purpose bearings. Electric motor service bearings are also listed and illustrated.

Clearing Cushions — Clearing Machine Corp., 6499 West 65th St., Chicago, Ill. Booklet, 20 pages, 8½x11 in. Improved type of air cushion for power presses with single, double or triple pistons. Illustrated with photographs and cross-section line drawings. Specifications for different designs and sizes are also given.

Controlled Rectifiers — Ward Leonard Electric Co., Mount Vernon, N. Y. Bulletin 8601, single sheet. For supplying d.c. output from a commercially constant, single phase, a.c. line of 110 volts. Gives sizes and power input and output.

Emblems, Badges and Miscellaneous Items — American Emblem Co. Inc., Utica, N. Y. Catalog, 74 pages, 6x9 in. A pictorial booklet covering the products made by this company. Although emblems, badges, buttons and similar parts are made, facilities include equipment for all types of press and drop hammer work. All types of metals are used for such items as metal boxes and containers, clock cases and bases, novelties, nameplates and escutcheons, and industrial stampings and products.

Diesel Engines—Caterpillar Tractor Co., Peoria, Ill. Booklet 13 pages, 4x9 in. Answers the questions "Why the diesel engine" and "How does the diesel run?" Compares the diesel to the gas engine.

Electrotinning — E. I. duPont de Nemours & Co., Inc., Wilmington, Del. Manual, 12 pages, 8½x11 in. A summary of the industrial features of the du Pont sodium stannate-acetate electrotinning process. Shows standard methods of preparing, controlling and analyzing the solution. Preparation of the base metal is discussed.

Gaskets—Goetze Gasket & Packing Co., Inc., New Brunswick, N. J. Catalog 48, 64 pages, 8½x11 in. A very complete book showing many types of gaskets, among which are included in cross-section 17 of the most popular gasket designs. Well illustrated with installation photographs and line drawings. Gaskets are listed for sizes up to 13 ft. diam. and for working pressures from 125 to 1,500 lb. capacities.

Nickel-Clad Steel—Lukens Steel Co., Coatesville, Pa. Bulletin, 24 pages, 8½x11 in. A pictorial and tabular presentation on the uses of clad steel. Physical, chemical and corrosion-resisting properties are given. Sizes, weights, forms, and fabrication are discussed. More than 100 materials tabulated include oils, acids, gases, juices, liquors and water as used in the various industries or processes. The application of clad steels to the containers used are also discussed.

Precision Needle Rol'er Bearings—Folder, 8½x11 in. Needle bearings for use where minimum outside dimensions, minimum weight, concentrated load capacities are requisite. Applicable to wrist pin end of connecting rods, cam rollers, rocker arms, pump shafts, gears, engines printing machinery and miscellaneous machine purposes. Sizes 12 to 150 mm. bore available. Table of sizes and load ratings included.

Pumps — Acme Machine Products Co., Inc., Muncie, Ind. Catalog 140, 23 pages, 8½x11 in. Self-priming, "Sure Flow" pumps for coolant, oil, water or other fluids filled with abrasives. Pump capacities up to 100 gal. per min. available with direct motor or belt drives for low and high-pressure duty. Either horizontal or vertical bases supplied. Dimensioned mounting outlines and cross-sectioned drawings included.

Rotary Converters — Janette Mfg. Co., 556-558 W. Monroe St., Chicago, Ill. Folder, 8½x11 in. For converting d.c. to a.c., these converters are used for radio receivers, power amplifiers, motion picture projectors, testing equipment, electro-therapeutic machines, gaseous electric signs and radios in airplanes. A table of sizes and weights are given.

Tachometers—American Schaeffer & Budenberg Div., Consolidated Ashcroft Hancock Co., Bridgeport, Conn. Catalog 500, 16 pages, 7⅞x10½. For indicating speeds directly in r.p.m. Horizontal drive with horizontal shaft, vertical drive with vertical shaft, vertical shaft for horizontal drive and flexible shaft drive are illustrated. Used with engines, turbines, printing presses, paper machines, conveyors, compressors, blower and similar machinery.

Valves—Hancock Valve Div., Consolidated Ashcroft Hancock Co., Inc., Bridgeport, Conn. Booklet, "How Long Will Its Heart Beat?" 8 pages, 7⅞x10½ in. Showing with photographs the test of a valve seat and disk of ordinary material compared to that of stainless steel with a Brinell hardness of 500. Also show cut-away sections of valve assemblies and gives a list of materials used for the parts.

Valves—Homestead Valve Mfg. Co., Coraopolis, Pa. Folder, 8½x11 in. Plug valves with lapped seats, protected seat poppet type valve with soft sealing disks, and seat and disk valves with regrindable, reversible and renewable seats and disks. Brass, semi-steel and cast steel valves available.

Valves, Pumps and Other Products—Hills-McCanna Co., 2349 Nelson St., Chicago, Ill. Folder, 8½x11 in. Pumping equipment for the chemical process industries. Diaphragm valves, centrifugal pumps, chemical proportioning pumps, and lubricating systems. Thermometer wells and pyrometer protection tubes are also shown.

Wheels and Casters—The Wellington Machine Co., Wellington, Ohio. Catalog 56, 28 pages, 7x10 in. Lists rubber tired wheels and casters mounted on solid roller, spiral roller, and tapered roller bearings. Integral rim wheels, wheel collars, special mounting brackets for casters, hangers and spring mountings are also shown.

BOOKS AND BULLETINS

Elements of Machine Design

Dexter S. Kimball and John H. Barr. 476 pages, 5½x9 in. 226 illustrations. Blue clothboard covers. Published by John Wiley & Son, Inc., 440 Fourth Avenue, New York, N. Y. Price $4.

As stated by the authors in the preface to this third edition, no radical changes have been introduced, but new subject matter has been added and some obsolete sections have been eliminated. In general, the form of presentation follows previous editions. Written primarily for use as a college textbook, the first chapters deal largely with definitions and fundamental principles.

However, it must not be inferred that the book confines itself to theoretical considerations. Indeed, a more than usual amount of space is devoted to explaining the practical limitations of the formulas and equations, and why the actual conditions will cause considerable deviation from the theoretical analysis.

In addition to detailed analyses of the design of bearings, journals, axles, springs, and various other machine elements, the text presents the generally accepted standards such as the A.S.M.E. standards for shafting, the recommendations of the American Gear Manufacturers' Association, standard ball and roller bearings, A.S.T.M. specifications and similar standards.

Finishing Metal Products

Herbert R. Simonds. 337 pages, 6x9 in. 147 illustrations, 7 tables. Blue clothboard covers. Published by McGraw-Hill Book Co., Inc., 330 West 42d St., New York, N. Y. Price $3.50.

This first edition was written to point the way to more attractive finishes and to more efficient means of producing these finishes. Besides many of the new finishes, the book covers the usual types such as galvanizing, polishing, plating, lacquering and painting. Because of the close alliance of the preparatory processes and the metal-finishing technique, several chapters are included on cleaning and descaling.

The book has six main divisions. In Part I—Commercial Aspects of Finish and Appearance the sales value of attractive finishes, selection problems, plating characteristics, importance of color and prefinished materials are discussed. Part II emphasizes the preparation of the metal for finish application and Part III tells about polishing and buffing. In this last mentioned chapter, not only are production problems discussed but consideration is also given to the technical side. Design suggestions of the product itself to reduce polishing costs are given. Fixture designs for buffing and polishing are shown in line drawings.

The subject of plating and metallic spray coatings is contained in Part IV. Nickel, cadmium, chromium and copper plating are discussed. Parts V and VI cover painting, enameling and lacquering and such finishes as porcelain enamel, alumiliting synthetics and novelty coatings.

Materials Testing and Practice

Irving H. Cowdry and Ralph G. Adams. 144 pages, 6x9 in. Clothboard covers. Published by John Wiley & Sons, Inc., 440 Fourth Ave., New York, N. Y. Price $1.75.

Dealing as it does with the fundamental stresses and strains and methods for their determination, this book is intended as a text to accompany a laboratory course, rather than an actual laboratory manual. With this in mind, the authors have presented basic principles which lend themselves to the widest application, omitting certain types of testing considered in the realm of specialists. Sections are included to acquaint the student with the commonly employed methods in the study of tension, compression, torsion, bending, hardness determination, shock and repeated stress.

This second edition, after a lapse of ten years since the first publication, includes some of the more important improvements and developments in testing technique, equipment and materials. A new chapter has been added on concrete testing and all specifications cited in the first edition have been revised. Some changes have been made in the text to meet difficulties encountered in the presentation of the subject matter to students during the past ten years.

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A Study of Vibration in Plant Machinery

The Felters Co., Inc., 210 South St., Boston, Mass. Booklet, illustrated, 32 pages 6x9 in. Paper covers.

Because of the advances made in noise and vibration isolation in automotive construction, this company has conducted laboratory and field tests for all types of machinery. Some of these tests, and conclusions developed from them, are published as a practical guide to executives and engineers interested in the subject. Wool felt is used as the isolation material. A concise chapter tells of the characteristics of wool felt. Another chapter discusses the isolation of vibration. Still another shows a study of wool felt placed under the legs of textile looms and illustrates the results with line diagrams. Noise isolation tests, conducted on a spinning machine, are recorded in another chapter with line drawings of the felt application to the machine feet.

Bibliography on Vibrations in Electrical Conductors

Published by the Power Transmission and Distribution Committee of the American Institute of Electrical Engineers, 33 West 39th St., New York, N. Y. 317 pages, 8½x11 in. mimeographed.

Includes all supplements and additions to December 1935 making a total of 2,352 titles of scientific and engineering articles on vibration and fatigue of cables. Nine sections include vibrations observed at low wind velocities, galloping conductors, effect of wind upon exposed circular or ice-coated bodies, wire rope research, sound and noise, fatigue of metals, water hammer and associated phenomena, and vibration dampers.

Gear Generators

Farrel-Birmingham Co. Inc., 344 Vulcan St., Buffalo, N. Y. Catalog No. 440, 80 illustrations, 52 pages, 8½x11 in. Paper covers.

Although called a catalog, the booklet is a condensed treatise discussing the theory of gear generation, with particular reference to the Sykes process. The design and construction, the set-up and operation, and specifications of the machines are given. A number of gear types and tooth forms that can be generated are described and illustrated; also included are sections on the lapping, measuring and testing of gears. A chapter of the Sykes continuous tooth herringbone gear shows a number of applications of this type of gear and discusses tooth contours, proportions, helix angle, efficiency, tooth contact and gear materials used. Diagrams are used to illustrate the principle of this process.

Cageless Roller Bearings

Tyson Roller Bearing Corp., Massillon, Ohio. Engineering manual, 106 pages, 8½ x11 in., mimeographed. Paper covers.

Third edition of this engineering manual contains pertinent data of the cageless bearing principle. Assembly of bearing, mounting, application, adjustment, lubrication and performance are discussed. Tables of fitting practices for automotive and industrial applications are given for both cup and cone. Dimension charts are drawn full size and can be removed from the book and used as templates when tracing. At the lower end of each sheet is given the r.p.m. from 100 to 5,000 with corresponding radial and thrust loads. Also an 18-page manual on mine and quarry car wheels gives recommendations for mounting with lip, labyrinth and combined nut and lip seals. Dimensioned line drawings show design details.